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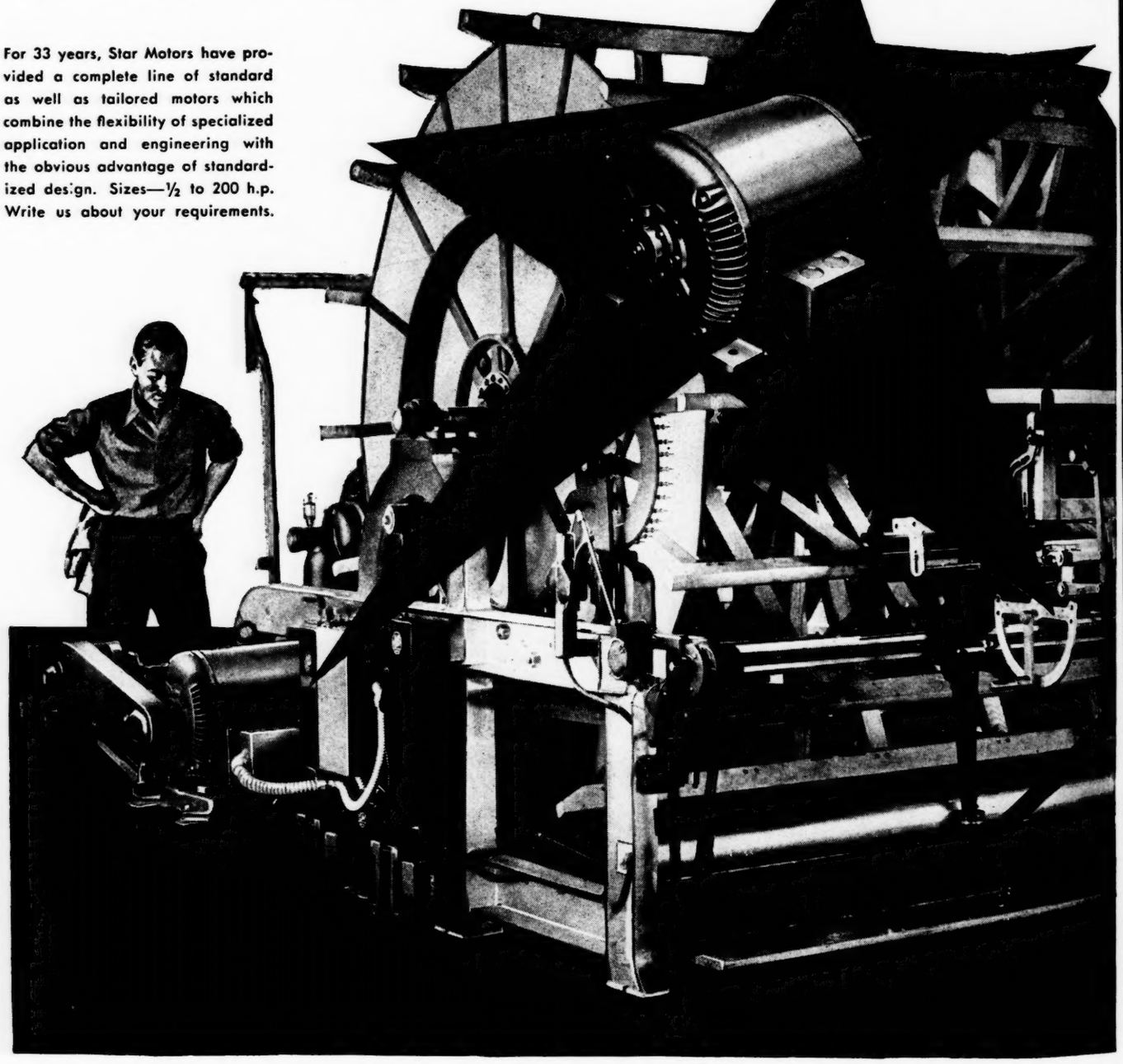

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HEATING AND VENTILATING

Publisher

THE INDUSTRIAL PRESS

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Publishing Information

HEATING AND VENTILATING is published monthly by The Industrial Press at 148 Lafayette Street, New York 13, N. Y. Western office, 228 North La Salle Street, Chicago 1, Ill. Yearly subscription in the United States, Spain, South America, Mexico, and Canada, $2.00; in all other countries, $3.00. Single copies 30 cents.

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New York 13, N. Y.

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Chicago 1, Ill.

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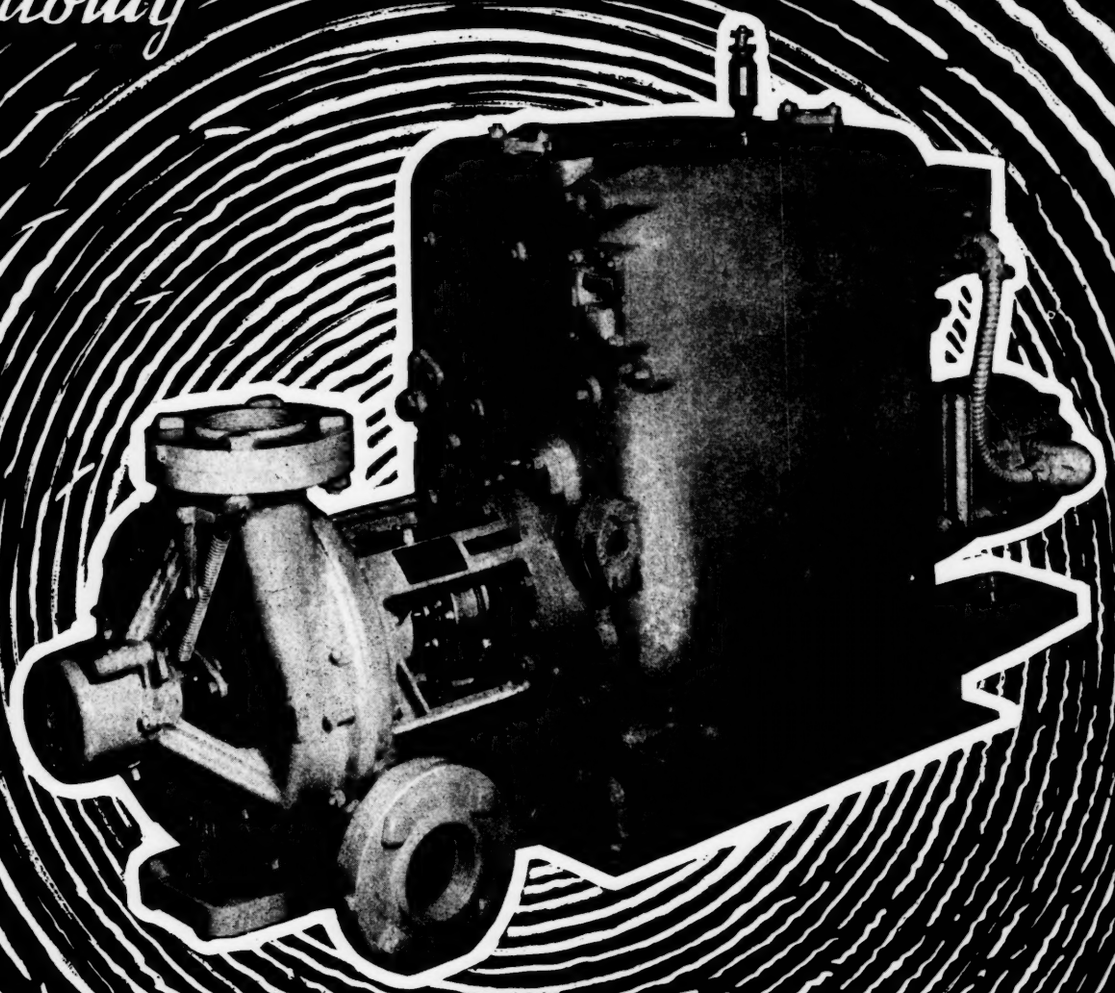
Next Month's Issue

A large convoy plodding along at night without lights excites the public's imagination. What it may fail to appreciate is that light-excluding ventilators and darkened portholes obstruct natural ventilation. "Blackout at Sea" will describe how ships are ventilated during the evening hours. The issue will contain another article on how refrigeration is being employed by industry to improve the quality of mechanical products.

Centralized Heating Economy



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HEATING AND VENTILATING

Fig. 1. Architect's conception of how one group of buildings of Clinton Hill Apartments will appear when completed. Only three buildings have been erected.

Apartments with Central Heating Plant

T. F. J. MOFFETT

Three apartment houses of a proposed group of eleven buildings, are being heated from a central heating plant through high vacuum systems provided with drop risers and convectors. Completed buildings of Clinton Hill Apartments house 310 families.

PLANNED, built, owned and operated by the Equitable Life Assurance Society of the United States, the Clinton Hill Apartments, Brooklyn, N. Y., stand out as a modern example of heating a group of apartments through a central heating plant.

While the Clinton Hill development contemplates 11 fireproof apartment houses 12 to 14 stories high, only three have been completed. The others will wait until materials are available. The structures are of reinforced concrete with brick exteriors. The completed buildings now house 310 families of Navy-yard and war workers. The apartments contain four rooms and five rooms and rentals include the gas and electricity used by tenants. The fact that electricity goes with the rent may be of interest to the ventilating industry. It invites liberal use of fans in warm weather and, as has already been demonstrated, use of a kitchen window fan plus the mechanical bathroom ventilation exhaust to clear an apartment of smoke when windows are almost closed and there is a social gathering.

The plot plan for six buildings is shown in Fig. 2. The three solid outlines indicate the structures already erected. A special city franchise was secured to permit future connections in conduits under the street to serve Building No. 1. The plot plan for the other five projected buildings is quite similar to that of Fig. 2. Structures have eight apartments to a floor and house from 96 to 108 families. The architects, Harrison, Fouilhoux & Abramovitz, and the firm of Edward E. Ashley, consulting engineer, have standardized the blueprint details so that a single bundle of drawings can serve for most of the details. Thus nine of the buildings will be known as Type A and two as Type B.

The three completed structures are close together and are joined by passageways in the substructure.

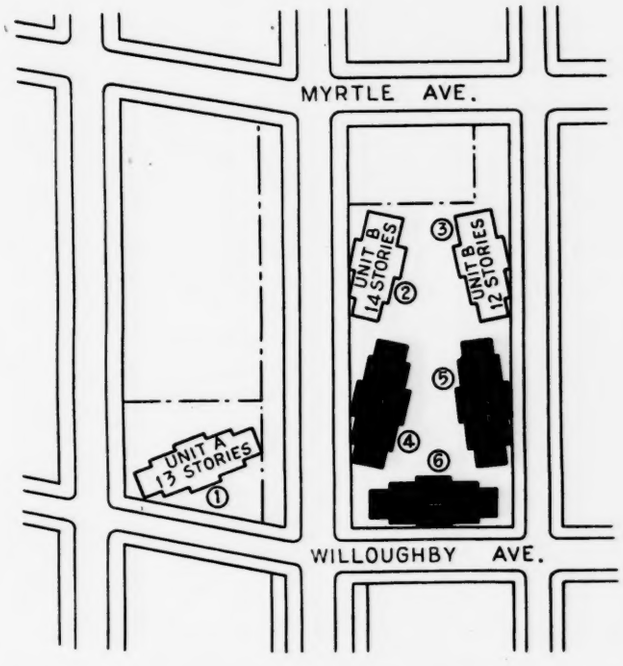

Fig. 2. Plan of the project. Buildings of one group to be erected are shown in outline.

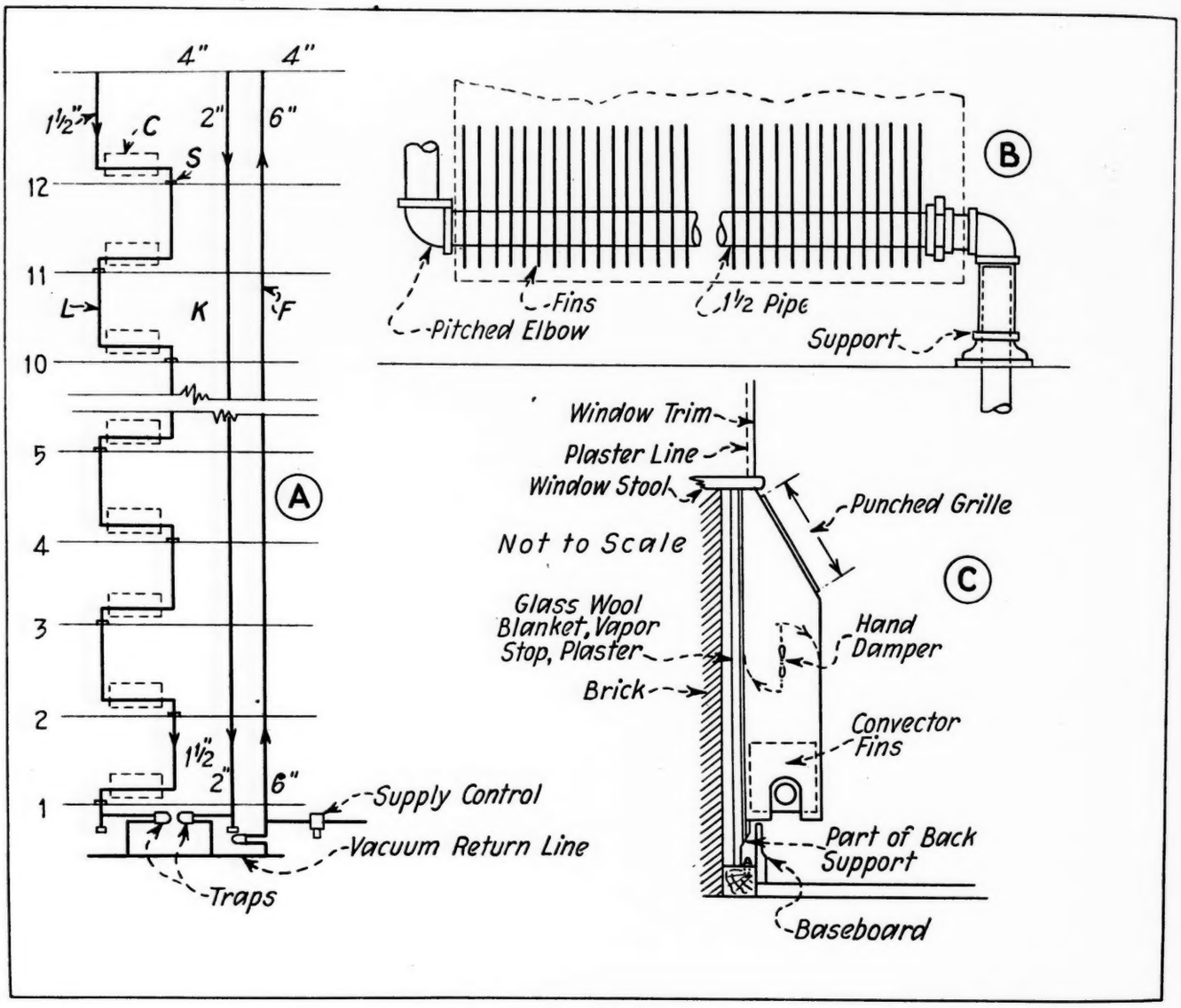

Fig. 3. Typical heating system details. Each building has at least 16 heating risers and 25 offset convector risers.

The present boiler room is in the westerly building, No. 4. It will eventually serve the three additional buildings when they are erected. The other group of five buildings will be several hundred feet distant and will have its own single boiler room. Piping plans provide that any of these buildings may be cut off and operated singly. Space is left in the floor plans for a chimney and shaft space for any required piping. Foundation walls have a bricked knockout panel which would permit bringing in a large boiler.

The present boiler room has four large steel firebox boilers, each of 270 hp capacity, and intended to carry the heating load of the six buildings. The boilers are set high over bricked combustion chambers, designed at first for oil firing, then considered for stokers with suitable coal conveying equipment. Because of material shortages, hand-firing was selected. Even during the coldest weather, two boilers will be sufficient to heat three buildings and supply steam for the hot water supply and the requirements of one restaurant. A single boiler can carry the load for the three buildings when the outside temperature is 40F or higher. It is not easy to dump and rebuild fires, however, so that in practice one boiler is really banked during mild spells. Three boilers will be sufficient for six buildings, thus leaving one regularly in reserve.

The heating system, installed by Baker, Smith & Co., contractors, combines flexibility and simplicity. It provides a spread of heating surface temperatures from that corresponding to 25 in. of vacuum up to several pounds gauge pressure, if required. The job is essentially a C. A. Dunham Company sub-atmospheric differential steam system, embracing the "Metro" method of risers and convectors for circulation.

Beginning at the boilers, the nozzle outlets connect into a 14-in. header from which branch supply mains run to each building. They are 8 in. for the three nearby structures and 10 in. for the more distant ones. The 10-in. mains are plugged for the present,

but eventually they will be carried underground in presealed conduit. The design starting pressure is 10 lb at the boilers. Line losses are entailed to the electrically operated differential control valve for each house heating system, and to the automatic valve supplying steam to the coils in the hot water storage tank.

In each building a single 6-in. upfeed riser carries steam to the distributing mains in the space above the top floor ceilings. Overhead mains connect to the various drop risers going down through bathrooms, kitchens and living rooms. All drip or float condensation traps are ranged along the basement ceiling, at the foot of each riser, and all are accessible for inspection without the need of entering any apartment.

Convectors in Living Rooms

Living rooms are heated by enclosed convectors placed under the windows. The design makes the convectors practically a mere section of each drop riser. The riser is staggered from floor to floor as it goes down, first on one side of a window, and then on the opposite. Risers are intentionally close to the edge of the window opening so that they can be concealed by window draperies. The riser is exposed as useful heating medium, and contributes something to the warming of the room. It is thus a gain over risers concealed within partitions or walls, although it might be argued that such risers heat indirectly by warming nearby wall surfaces as panels.

Primarily the radiator system at Clinton Hill Apartments was not designed to have the drop risers serve for useful heat. The real objective, when this method was first developed by H. G. Shaefer, heating engineer, was to attain high installation efficiencies through a standardization of piping parts and a simplicity of required make-up. Aside from the convector section, only one piece of piping is needed to a story. Riser lengths may be cut in advance to standard size.

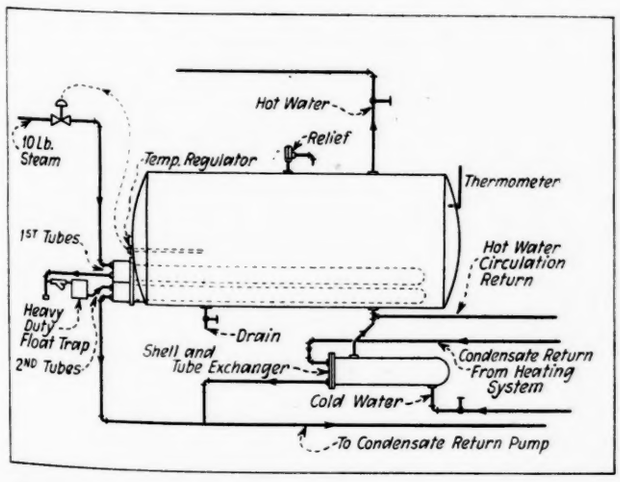

Fig. 5. Diagrammatic layout of heat recovery connections of condensate return lines at hot water storage tanks.

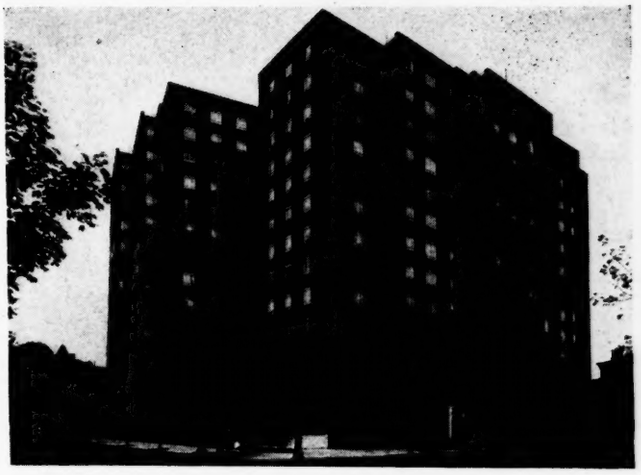

Fig. 4. Two of the completed apartment buildings.

This method of steam heating circulation is shown in Fig. 3-A. The 6-in. upfeed riser *F* starts at the supply control valve. Overhead mains branch around in the space under the roof. The drop risers *K* for kitchens and bathrooms are 2 in. and run almost straight down to drain out at the bottom through float and thermostatic bleeder traps. These vent air along with the condensation. To provide for expansion in the risers, they are anchored about midway up in the building. Branch connections at both top and bottom have sufficient swing. Each building has about 16 heating and 25 convector risers.

Risers dropping to the convectors *C* are 1½ in. all the way and connect ordinarily with a single piece of pipe from one floor to the next. The horizontal convector sections within the enclosures have fins attached to the pipe and the whole makeup is about as indicated in Fig. 3-B. The fins are 4¾ in. wide by 4 in. They are made of sheet steel attached mechanically to the pipe by well pressed interlocking joints which have good contact area. The core of the convector is regular pipe and could serve high pressures. Elbows for making the offset at each floor are pitched to an angle of at least 91 deg. instead of the standard 90 to assist gravity drainage.

In Fig. 3-A an ordinary length of riser is marked *L*. Assuming a standard height to the stories, these lengths could be cut to measurement. One end of the convector rests on a floor support. The point of support is marked *S* and is required at each floor. There is then ample leeway for expansion in the line between the supports at each story and other anchors or hangers are not needed.

Remembering the objective of installation efficiencies, it will be noted by practical heating men that the system of Fig. 3-A almost falls into place once the sleeves have been properly located in the concrete floor slabs. These sleeves usually are of seamless tubing, 2½ and 3 in. OD for 1½ or 2-in. pipe risers, respectively, and formed so that wall extrusions serve to anchor the pipe in the concrete. Since sleeve openings are free of the baseboard, it is not necessary to put in the pipe lines before the

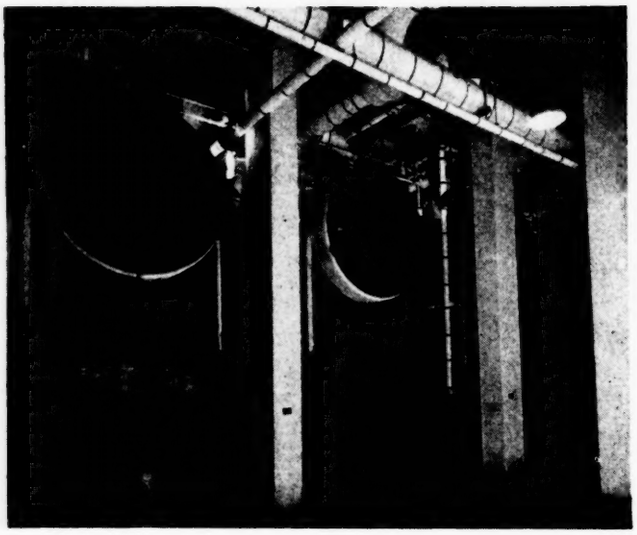

Fig. 6. The central boiler room has four Pacific steel boilers ready to serve six buildings. While hand firing is used at present, stokers can be installed, or oil burned.

walls are plastered. This keeps lime and other accumulations away from the convectors. A supporting wall panel at the back of each convector enclosure may be installed before the risers.

At the foot of each riser a 1-in. float and thermostatic trap takes care of draining and venting. All traps connect to the vacuum return line. There was a time when good heating men expressed misgivings concerning the amount of condensation which would accumulate in the lower convectors and would interfere with proper heating or smooth operation. Service tests have shown no trouble of the kind.

Improved Type Convectors Used

The sheet metal convector enclosures at Clinton Hill are of improved design, particularly at the top grille outlet, and extend into the room less than 5 in. The flat enclosures in other installations had been used for storing newspapers and magazines. The new pattern has a sharply slanted top to prevent such cluttering. The enclosure front may be lifted off after turning a key-operated eccentric hook or lock. Heat to a room may be controlled by setting a hand damper over the convector. When the damper is completely shut, the heat output of the unit is reduced about 75 per cent.

Heating engineers are acquainted with the Dunham system of differential sub-atmospheric steam heating. The controlled variations in vacuum pressures regulate the amount of heat given off. At Clinton Hill, the system anticipates the use of gauge pressure, if ever required, and the available system temperature range is approximately 85F. This would be from about 135F at high vacuum up to about 220F with 3 lb gauge.

Dunham equipment installed thus far embraces two duplex vacuum pumps, one serving two of the buildings, and the other a single building. The larger pump is rated for 40,000 sq ft of radiation and the smaller one for 15,000. A differential controller for each system regulates the vacuum pump operation. The controller automatically starts the vacuum pump when the pressure difference has narrowed considerably and the steam circulation is thus accelerated. The effect of vacuum pump operation is as much for active steam circulation as for holding a partial vacuum.

Panel Set for Each Building

A central control panel or central operating station for each building is fitted with indicating lights, mounted three in a row on the boiler room wall and interconnected electrically for master control. The flow of steam within the differential heating system is automatically regulated by a selector and balancer, the selector being in effect a thermostat responsive to outside temperatures but mounted on the inside glass surface of an exposed window. The heat balancer uses a test radiator and serves to measure the heat output and its rate of supply in a manner to insure that heat supply is in keeping with demand. Between them, these two controls actuate the electrical balance control within the panel which in turn starts a relay for opening or closing the control valve. Various timing and cycling arrangements may be made for automatic operations such as controlling the steam supply so that the entire load is not simultaneously imposed on the plant in the morning when starting up the heat for three buildings.

A few random radiators at Clinton Hill use regular packless supply valves and standard radiator return traps. These are in vestibules, lobbies, corridors, elevator machinery rooms, and in spaces reserved for stores. They are connected by branches or risers to steam mains on the boiler side of the differential control valve. Return lines run separately to vacuum return pumps. Returns from coils in the hot water

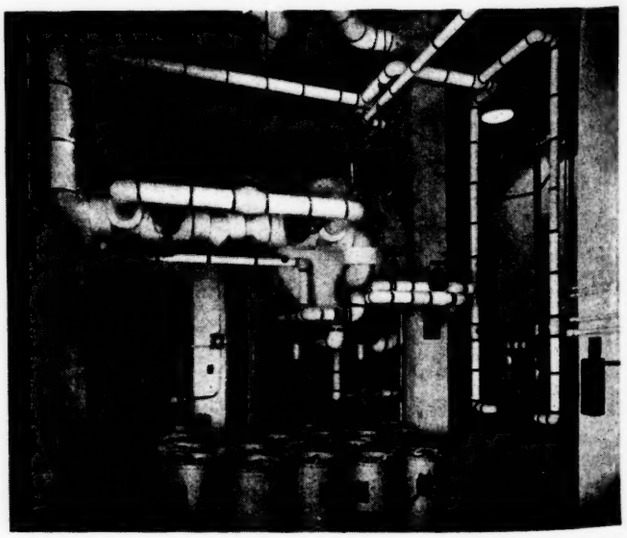

Fig. 7. Section of the boiler room showing hot water storage tank and connections for one of the buildings. In recess at left is the ash can lift. At right, is the vacuum heating system pump for two structures.

Fig. 8. Dining room of representative apartment with draperies held back to show setting of drop riser and convector enclosure. Convector damper handle is at right side of enclosure.

storage tanks connect to a small receiver and condensation pump in each building. These pump directly through the condensate return line to the boilers.

The cement lined hot water storage tanks hold 1,400 gallons. They have thermostatically controlled steam valves which carry the full available boiler pressure. The heating elements in the tanks are multiple type removable U-tubes in two coils set one above the other. The Ashley engineers have arranged an economizing method by which condensate from the upper coil passes through its float trap and goes out through the lower coil, giving up most of its retained heat on the way.

This arrangement is outlined in Fig. 5 from a layout of typical blueprint details. Clinton Hill does not as yet employ the shell and tube exchanger for recovering heat from the heating system returns because the present buildings are near the boiler room, but the method will be available for more distant buildings where condensate heat may be utilized before it cools off in long returns. The incoming cold water passes through the exchanger and absorbs any available heat.

At the job, operating superintendent James Bazemore, a former marine engineer, has been in charge for several months. He took over a new plant and has organized it into smooth operation. Buckwheat anthracite coal is being burned and consumption during cold weather appears to average about 5.5 short tons a day. For 310 families this would amount to about a bucket per day on the average for heat and hot water per apartment, assuming about 26 lb coal per bucket. The old cold water flat with kitchen range and potbelly parlor stove did not do as well in economy of fuel.

The heat is turned down over night at Clinton Hill from 11 p.m. to 5 a.m. However, the banked fires continue to maintain hot water supply. Exhaust fans are also shut off over night. These include eight high duty roof type ventilators for the eight bath room shafts in each building. Fans are controlled by a master switch in the boiler room, and at a control panel with individual switches and signal lights located in the penthouse. Ventilation is also provided for the elevator corridors at each floor with a grille outlet near each end and a system of ducts leading to a large roof type exhaust fan, without the standard housing in a louvered enclosure on the roof.

The mechanical ventilation embraces a liberal system of basement heater room and toilet room exhausts using propeller type exhaust fans. Balanced dampers permit fresh air to enter the heater rooms when the exhaust fans are operating. Louvered openings in doors permit cross ventilation through basement areas. Other exhaust fans will take care of cafeteria kitchens, future boiler room, employes' rest rooms, etc. Exhaust fans are installed also for ventilating the air space between the hung ceiling in the boiler room and the first floor, and also to prevent boiler room heat from causing some discomfort on that floor.

DESIGN AND CONSTRUCTION

Architects—Harrison, Fouilhoux & Abramovitz.
Consulting Engineer—Edward E. Ashley.
General Contractor—Starrett Brothers & Eken.
Heating Contractors—Baker, Smith & Co., Inc.
Plumbing Contractors—J. L. Murphy, Inc.
Resident Engineer—James Bazemore.

New Standards Group to be Formed

Organization of an Allied Nations standards body was announced at the silver anniversary meeting of the American Standards Association. Affiliates include groups directly and indirectly related to the heating and air conditioning industries.

Objective of the new body will be maximum coordination of standards necessary for the war effort and the immediate postwar period. A skeleton staff will have offices in London, and in either New York or Washington.

R. E. Zimmerman, vice-president, United States Steel Corp., was succeeded as President of the American Standards Association by Henry B. Bryans, Executive Vice President, Philadelphia Electric Co. Reelected were: Vice-President, George S. Case, Board Chairman, Lamson & Sessions Co.; Chairman of the Standards Council, H. S. Osborne, Chief Engineer, American Telephone and Telegraph Co.; Vice-Chairman of the council, E. C. Crittenden, Assistant Director, National Bureau of Standards.

A telegram from Donald Nelson, WPB Chairman, declared that "applied standardization must be one of our most important means of expanding output and achieving more efficient mass production."

Irradiation as a Health Factor in Air Conditioning

F. W. HUTCHINSON

Assistant Professor of Mechanical Engineering,
University of California, Berkeley, California.

TWENTY-THREE hundred years ago, in a book ("Meteorologica") that is as entertaining to the engineer of today as it was instructive to the natural philosopher of that time, Aristotle recorded a theory of radiation which, though incorrect, is interesting in that it reflects ideas which have not yet been entirely abandoned by the layman. Aristotle believed that "sight" is a form of radiation emitted by the human eye in all directions; a radiant stream passing from the observer to the observed object from which it is reflected back to the eye. But how, we ask, could such a fantastic theory continue to be accepted, lacking as it must be in experimental verification? The answer, as always, is that few firmly established theories long remain lacking in evidence which can be adduced as experimental "proof." To substantiate the hypothesis in question Aristotle cited the case of a man whose sight was so weak that it could not pass even through thin air and was therefore reflected back to him; thus wherever he walked he was troubled by a reflected image of himself which was perpetually approaching him.

Self-Irradiation

The Aristotelian theory of reflected "sight" has long been abandoned, but the idea of self-irradiation, that is, absorption by the body or clothing surface of radiant energy initially emitted by the person, is still of sufficient interest and practical importance to have been the subject of a recent research paper published by the American Society of Heating and Ventilating Engineers. In this paper tests were reported indicating that conditions of optimum comfort can be realized in a room having an air temperature of 58F if the enclosure is surfaced with a material having a high reflectivity. Thus radiant energy emitted by the human body does play an important part in establishing that physiological heat balance between energy production and energy dissipation which is so vitally important to the realization of comfort and of health.

However, self-irradiation of an occupant in an enclosure with surfaces of the usual type, that is, having low reflectivities for radiation in the infrared, occurs only in an insignificantly small amount; the greater fraction of irradiation is of energy emitted by surrounding surfaces, received through windows, or received from flame, as from a residential fireplace or industrial furnace. From the standpoint of health, therefore, the principal interest in irradiation is with respect to that fraction of the radiant energy received directly by the human body which differs appreciably in quality from the energy being emitted by the body and the clothing surface. In this respect the health influence of low temperature radiant heating or cooling systems—as panel heating—would seem to be important to health not in terms of the direct effect of irradiation, but indirectly due to the influence of the net radiant exchange on the body heat balance.

Radiant Exchange as a Comfort Factor

The effect of radiant exchange as a heat balance factor in establishing an environment having satisfactory comfort and health characteristics is basically a problem of satisfying the heat transfer and heat balance equations and therefore can be treated in the same manner as temperature or air motion. Thus the external factors of temperature, humidity, air motion and radiant exchange combine with the internal factor of metabolic rate to establish a heat balance on the body which is, or is not, comfortable and healthy. The influence of each of these factors can be evaluated independently of the special conditions such as drafts, clamminess, unbearable intensity of radiation, which may lead either to discomfort or to unhealthful conditions (or both) even though the basic heat balance is satisfied.

Specifically, then, the direct health effects of irradiation can be evaluated independently of the complete environment and in terms either of the quantity of irradiation, for intensity effects, and the quality of irradiation for effects due to absorption of energy from high temperature, or otherwise unusual, sources.

Quality of Radiation

The spectroradiometric curve has been generally accepted as the most effective means of expressing the quality of thermal radiation emitted by a source. This curve, a plot of quantity of energy as a function of wave length, permits immediate visual evaluation of the source characteristics in terms of the effectiveness of irradiation (in killing bacteria, producing erythema, causing fluorescent excitation or contributing to any of the other effects which result

when radiant energy is absorbed). Use of the curve also brings out the important fact that the specialized and very different effects resulting from absorption of radiant energy are occasioned entirely by the characteristics of the receiver; all radiant energy is fundamentally subject to the same analytical treatment, differences in effect being due solely to variation in the wave length.

For all except the most unusual air conditioning problems, interest is limited to sources for which the quantity and quality of the emitted energy is determined by source temperature; energy from such sources is called *thermal radiation* to distinguish it from the characteristic selective radiations such as occur in a mercury arc or a neon lamp. Thermal radiation is characterized by two highly significant properties: 1. Emission occurs over a wide range of wave lengths, but the spectroradiometric curve is smooth and continuous. 2. The quantity of energy emitted increases with the fourth power of absolute temperature and mean wave length decreases as the temperature increases.

Experiment has established that the germicidal effect of radiant energy is limited to that section of the ultra-violet region between (roughly) wave lengths of .2 and .3 microns, (1 micron, μ, is equal to 0.00003937 inch). The erythemal or tanning effect of radiation is likewise limited to a very narrow band of wave lengths; maximum erythemal effect apparently occurs at .297 microns while outside the range from .285 to .310 microns erythemal effect rapidly falls off from 10% of maximum to zero. The human eye is sensitive (in terms of light) to radiant energy between wave length of .4 and .7 microns. Wave lengths greater than .7 μ are used in drying processes, heat-therapy and panel heating; very long wave lengths, over 100 μ, correspond to radio waves and, over 10,000 μ, to energy waves from an electric generator. At the other extreme, X-rays, Gamma Rays (from radium) and Cosmic Rays appear successively as the wave length is decreased. The known range of wave lengths is so great that if a scale were chosen on which the visible region occupied a band 1" wide, representation of electric waves would require extension of the scale to a length equal to 8 round trips to the moon.

In spite of the enormous spread of wave lengths which is indicated above, the radiation significant in air conditioning problems usually does not depart very far from a relatively narrow band. This is due to the fact that the two most common high temperature radiant sources, the sun and the common tungsten incandescent lamp, pass through filters before reaching the occupant. After passing through the atmosphere only 1.3% of solar energy (on a clear day) is biologically effective; likewise such ultra-violet as is produced by a tungsten filament is almost entirely absorbed by the usual type of glass bulb. By enclosing tungsten filaments in fused quartz or in special types of glass some ultra-violet transmission can be obtained.

Health Effects of Infra-Red Irradiation

Usual heating sources (as radiators, low temperature panels, etc.) operate at temperatures so low that practically no radiant energy is emitted at that wave length. If this were not true such sources would be visible in a darkened room. But the spectroradiometric curve shows that wave lengths which are biologically effective, the ultra-violet, are even shorter than those which are responsible for light; thus no object which is insufficiently hot to be visible in the dark can possibly contribute radiation which will have any germicidal or erythemal effect. This elementary fact is obvious, from the spectroradiometric curve, but seemingly is not clear to many laymen who continue to confuse the effects of low temperature radiation with those of high temperature radiation. Such confusion can, in part, be explained by the claims which have not infrequently been made that "radiator heat is identical with sun-ray heat"; beyond doubt the heat finally resulting from absorption of radiant energy is independent of the wave length, but the characteristics of the energy itself and its primary effect on the receiver may vary very greatly with wave lengths. Thus neither a cast iron radiator, a heating panel, nor any other low temperature source emits energy of anything like the quality of solar radiation.

Not only is the radiation from a radiator unlike that from the sun, but it is unlike that from other commonly used sources. The long standing "nose-opening" controversy carried on by Sir Leonard Hill and other investigators is based on the difference in physiological effects of the kind of radiation received from dark (long wave) and bright (short wave) radiating bodies. Hill's work has led him to believe that the stuffy uncomfortable feeling experienced in a closed warm room is due to the action of long wave length radiant energy on the skin and, reflexly, through the nerves of the mucous membranes, on the nose; seemingly there is a congestion and closing of the nasal passage. Hill finds that when the dark source is replaced with a light source (as a tungsten arc) the congestion is relieved and he observes a "nose-opening" effect and an increase in respiratory movement.

Von Dishoeck substantiates Hill's findings that the nose-opening, although usual, is not found in every case, while other investigators claim that the nose-closing effect is due only to heat and not to specific wave lengths of radiation; they fail to observe any nose-opening effect. In a reply to his critics Hill states that their inability to observe the nose-opening effect is due to lack of precision in measurement; this comment suggests that the controversy is not one with which the engineer need concern himself, for if the effect is so small that trained laboratory workers have difficulty in observing it, the possibility of its being of any great physiological importance is relatively slight.

In respects other than nose-opening, however,

radiation is important physiologically. Recent Italian investigators have found that the absorption of radiant energy produces a hyperemia of the inner organ tissues and in doing so raises physiological tone. They compare this effect with the hyperemia of the skin and relative anemia of the inner organ tissues which is known to result from convective heating and conclude that convective heating lowers the tone of the body and invites bacterial growth, while radiant heating improves the tone; they submit statistics showing that the incidence of grippe is an inverse function of exposure to radiant heat. Other writers have shown that exposure to irradiation intensities of 4 to 5 calories per square centimeter of skin surface per minute (chiefly from the red spectrum) causes a rise in body temperature (.7F), improvement in heart contractions (36 to 44), and increases the respiratory count (53%), lung ventilation (60%), and oxygen consumption (from 22% to 35%); the energy expenditure was increased 4.5%.

An investigation in Italian factories showed that the incidence of grippe and other respiratory diseases is less among workmen who are exposed to radiation. Additional evidence in support of the belief that infra-red irradiation is healthful appears from the results of artificial irradiation of Swedish coal miners for 3 minutes every second day over a two-year period; decreases were observed in one-day illnesses and in catarrhal diseases of the respiratory tract. The evidence from this latter case is somewhat dubious, however, since ultra-violet irradiation accompanied the infra-red and the importance of the infra-red alone cannot, therefore, be evaluated. On the basis of such data as are now available the qualitative effectiveness of infra-red seems to be well established, but quantitatively its effects are not well known. Hollander recommends using infra-red radiation in the medical treatment of acute upper respiratory infections.

An unusual case involving an undesirable effect of infra-red exposure was reported recently (1942) in the *Journal of the American Medical Association.* After five minutes exposure to radiation from an ultra high frequency transmitter, a man experienced a sensation of numbness throughout his body. As soon as exposure ceased the numbness disappeared, but a feeling of inebriation replaced it and persisted for over half an hour.

Health Effects of Ultra-Violet Irradiation

German investigators have found that ultra-violet radiation tends to increase metabolism by excitation of the ends of the sympathetic nerves; infra-red, on the other hand, decreases metabolism by stimulating the parasympathetic system.

In connection with the absorption of radiation by the human body, it is interesting to note that the ability for selective absorption which is evidenced by the skin for short wave radiation (light) does not hold for the longer wave for which, regardless of color, skin is a perfect absorber.

The effectiveness of ultra-violet irradiation is known to be a function of exposure time as well as intensity. Thus the tanning effect of solar energy depends, approximately, on the product of erethymal intensity and time. The curing of rickets, for example, requires a daily dosage amounting to 1.5 erythemal watt-seconds.

The two most significant ultra-violet irradiation problems of immediate practical significance are probably those related to arc welding and to radiant disinfection. The welding problem now seems to have been effectively and conclusively limited to eye and skin effects; available data indicate that there are no determinable radiation hazards as far as systemic effects are concerned (National Safety Congress Transactions, 1942). An indirect welding hazard may exist as a result of artificial ionization.

The bactericidal action of ultra-violet radiation as an aerial disinfectant has been well established, but its effectiveness for this purpose in ordinary air conditioning installations remains subject to question. The destructive action on bacteria and on air-borne viruses depends on the irradiation of the organism rather than on the radiation from the source. Thus, just as in other radiant exchange problems, effectiveness depends on the shape factor of the average organism with respect to the source (thus varies with position, size, orientation and distance of the organism) and on the rate of energy absorption by the intervening atmosphere (thus depends on absolute humidity). In addition, the type of organism will influence the lethal effectiveness of the irradiation while the quality and strength of radiation from the source will, for a given shape factor, determine the irradiation. As in all radiant exchange problems, a first requirement is that the distance separating source and receiver be kept as small as possible; application of this principle to use of ultra-violet as a germicide suggests that a "curtain" based on a vertical source (as for a hospital operating room door) would be more effective than one from a horizontal source across the top of the door.

In spite of the positive germicidal action that has been obtained in many experimental installations, present opinion seems to be that ultra-violet as a disinfectant has greater future possibilities than it does present realities. Bedford, discussing its application in present day (1943) England states that ultra-violet is, in general, of dubious value—particularly when used in dusty air. However, he suggests that it may be of value in such crowded and improperly ventilated spaces as air raid shelters. Coblentz has reviewed the literature (December, 1942) and concludes that the use of ultra-violet as a disinfectant "is not justified and if not used intelligently . . may cause serious injury to the occupants of places undergoing disinfection." •

(Continued on page 107)

How to Burn Coal in Large Buildings

IRWIN R. HOFFMAN
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THE present fuel crisis has focused attention on stricter coal economy and on the use of bituminous to replace anthracite wherever possible. With possible wider use of soft coal to help alleviate suffering, particular attention will have to be paid to methods of reducing smoke. While bituminous coal is not available in sufficient quantities to ease the shortage, it is more plentiful than anthracite.

It is general practice today to have an anthracite fire three or four times as thick as the average piece of coal. As the ash builds up, the fire will be higher; however, the fuel bed itself should follow this rule. Over 30 years ago, Professor Thurston recommended a fire five times as thick as the coal. In those days pea coal was the predominant steam fuel. The tendency since then has been to small anthracite with thinner and more efficient fires.

For best results, the furnace should be charged lightly and often. Care must be taken never to agitate or molest an anthracite coal fire except when cleaning. Since hard coal deadens quickly if disturbed while burning, fires must be cleaned as rapidly as possible. The fuel bed should be maintained uniform. Burned out spots can be covered by leveling or with the shovel if the spots are small.

Drafts are regulated in accordance with steam demand, and the damper opening kept at the absolute minimum. If this is not kept in mind, operators will emulate the practice in southern orchards of heating the atmosphere, but with no beneficial results in heating plants.

When clinkering anthracite is encountered, the difficulty of burning it can be lessened by running a bar along the top of the grate just after cleaning, when the first layer of coal has ignited. If this is done carefully, the fire will not be disturbed and any clinker that may be forming will be loosened. At the same time, this operation will tend to keep open the air spaces in the grate. The tool can be 12 to 15 in. long and not over ⅜ in. thick, attached as a cross piece to a convenient length of pipe. Clinkering can also be reduced and economy increased by blending soft coal of desired characteristics with small anthracite. The fuel bed is thus made more porous and air is uniformly distributed through it, aiding better combustion.

For burning soft coal the methods are altogether different. Soft coals cake together and air will only pass through the breaks and cracks. If undisturbed, the coal will burn out rapidly in these places. Too much air will flow through the fire, thus cooling the boiler and retarding steaming. Therefore, bituminous coal fires must be broken and levelled, to loosen the coke formation and to keep the burned out spots covered. Care must be taken not to turn the fire over as this brings ash to the surface where it melts and causes clinkering.

One of the best ways of handfiring soft coal is by the coking method, if the load on the boiler is not too heavy. Fresh coal is placed on the dead plate and the front portion of the grates. The gases distilled from the coal pass over the fuel bed and burn. When the coal appears coked, it is spread over the

(Continued on page 106)

How to Design Exhaust Hoods

By J. M. DALLA VALLE

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Part 12—Examples of Calculations Required in Designing Hoods

IN earlier articles of this series, it has been shown that control of air borne contaminants depends on providing an airflow pattern at their source of generation so that these contaminants are removed from the breathing zone of the worker. Thus, once the characteristics of openings or hoods are understood, and accounted for in design calculations, an exhaust system will perform as desired. Not only does a knowledge of opening and hood characteristics simplify design problems, but more efficient and economical systems are possible than by rule-of-thumb methods. However, a study of opening characteristics alone is not sufficient to assure good control. Coupled with it is the need to study the manner in which the contaminant is produced. This was discussed in Part 4 (June, 1943, page 91) of this series dealing with hood criteria.

While there are many ways of achieving control in any given instance, as a rule one method will prove better than the others if it (a) requires less air, and (b) offers the least interference with the performance of the workers' duties. In general, the best method selected will take into consideration the source and nature of the material controlled, and this will at once limit the kind of design which can be applied. When these factors are determined, the controlling air velocities will fix the air volumes needed. The overall design of the system proceeds in the usual manner.

In this concluding article are discussed methods of applying the information presented thus far. A few examples of practical interest will be given to illustrate the procedure.

Example A. At the point *A* (Fig. 85) is a source of dust which requires a controlling air velocity of 200 lfm. A hood 4 in. x 8 in. can be located only as illustrated, the point *A* being in the center plane of the hood. Determine the air volume the hood must handle in order to obtain an air velocity of 200 lfm at the point *A*.

1. The side-ratio of the hood in question is 1:2.
2. Refer to Fig. 11, Part 2 (April, 1943, page 46). Point *A* of problem is located 8 units outward and 5 units below the hood axis.
3. The contour value passing through point *A* is 12% of the air velocity at the opening.
4. 200/0.12 = 1750 lfm = velocity at opening.
5. Air volume through opening = 1750 × 4 × 8/144 = 390 cfm.

Example B. A tank 6 × 10 ft is provided with a canopy hood 8 × 12 ft. An air velocity of 100 lfm is required to control vapors coming from the liquid surface of the tank with an exhaust fan having a capacity of 20,000 cfm after accounting for duct resistance. How far above the tank must the hood

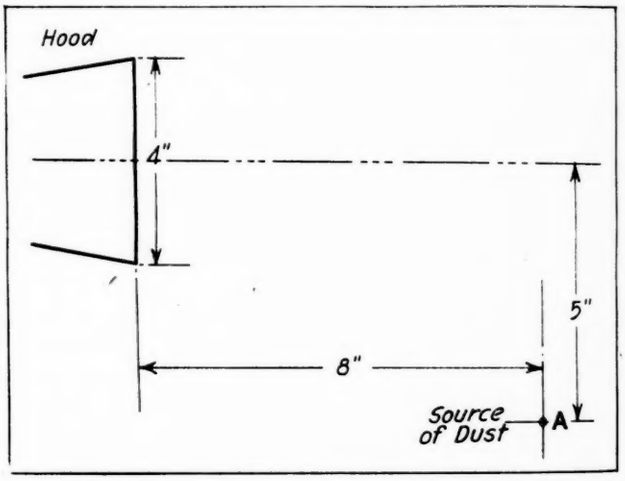


Fig. 85. Location of dust source for problem A.

be located in order that the controlling air velocity will be provided?

1. Use equation 25, Part 3 (May, 1943, page 86).
$$Q = 1.4\ PDV$$
Where Q is the air volume in cfm, P, the tank perimeter in ft, D, the distance above the tank in feet and V, the controlling air velocity about the tank in lfm.
2. Substitute the data given in the problem in the above equation,
$20{,}000 = 1.4 \times 2\ (6 + 10) \times 100 \times D$
from which
$$D = 4\tfrac{1}{2} \text{ ft.}$$

Example C. A vertical drill press is used for smoothing burrs on the ends of small pre-fired ceramic tubes. The tubes are approximately 3 in. length and ¾ in. in diameter. The burring operation reduces the length of the tube ½ in., and frequently causes chipping. The press turns at 500 rpm so that the material burred is thrown off with some force. The burring tool is 1 in. in diameter. Design a suitable hood to control the dust and chips that are present.

1. Tangential velocity of dust and chips leaving the burring tool = 1/12 × 500 × 3.1416 = 131 lfm
2. Assume largest chip weighs 0.1 oz. Therefore, energy involved, where W is the weight of the chip, is

$$E = \frac{Wv^2}{2g} = \frac{0.1 \times 131 \times 131}{16 \times 32} = 3.3 \text{ ft lb}$$

3. The energy must be counteracted by the force or pressure of the air on the chip and will vary as the square of the air velocity and the area of the chip in question, which in this case we shall assume to be ½ sq in. Using the formula for pressure on surfaces due to air velocity,

$$p = 0.00013\ v^2$$

where p is the pressure in lb per sq ft and v is the air velocity in lfm. Thus, for ½ sq in. the pressure is

$$p' = 0.00013 \times 0.5 \times v^2\ 144 = 0.00000045\ v^2 \text{ lb}$$

4. Now assume that this pressure must act in a space of 2 in., that is, it must dissipate the energy of the moving particle in a distance of 2 in. Hence

$$0.00000045\ v^2 \times 2/12 = 3.3 \text{ ft lb}$$

and solving for v

$$v^2 = 0.44 \times 10^8$$

v = 6600 lfm = air speed required to counteract projected particle in a space of 2 in. This velocity must be provided at the control point if there is no guard. If a suitable guard can be designed for the larger chips, then a control velocity of 4000 lfm may be deemed sufficient.

5. A suggested design is shown in Fig. 86. This design permits room enough for the insertion of the hand to set the pieces to be burred, and includes a guard to confine the chips to as small an enclosure as possible.
6. The estimated entrance area of the hood when the hand is holding a tube in place is 8 sq in.

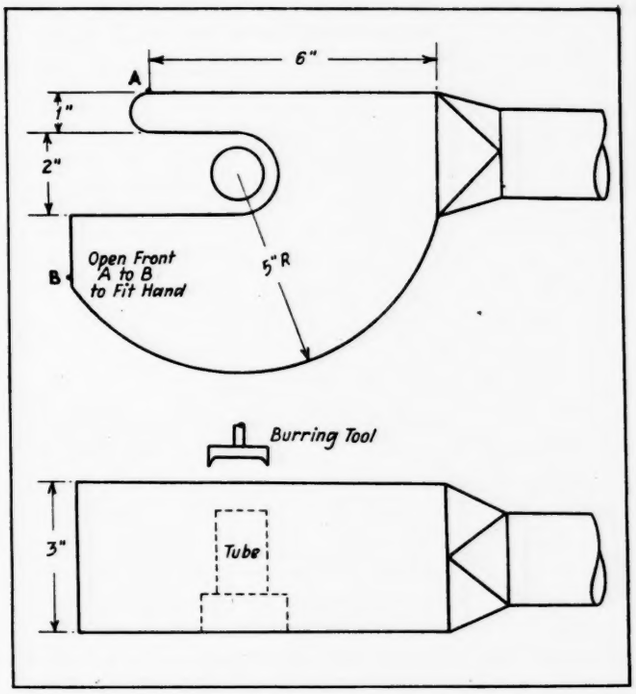


Fig. 86. Hood design for burring pre-fired ceramic tubes described in example C.

Hence volume of air required is

$$Q = 4000 \times 8/144 = 220 \text{ cfm.}$$

Example D. Metal rods 2 in. in diameter and 18 in. in length are to be "built up" by means of metallic spray methods. The jet and air pressure used provide for an air velocity estimated at 7000 lfm (about 100 cfm). The tip of the jet is to be held about 18 in. from the rod, and the velocity of the air stream at the work surface is 5000 lfm. The rod is to be rotated in a lathe, so that the hood designed must rest on the lathe bench. If the work area is enclosed, design a suitable hood for the job and estimate the air volume required for control.

1. The air jet will induce an air movement of the following amount: If the jet air volume is 100 cfm of free air, then according to McElroy, at 18 in. or 1.5 ft, $Q = 100\sqrt{1.5} = 120$ cfm.
2. The amount of air to be handled by the hood to remove the jet air is 100 + 120 = 220 cfm. In addition, the hood must control rebound of particles so they are kept within the zone of influence of the hood. Particles of molten metal will possess a velocity much in excess of the air velocity at the work surface. An approximate estimate is 5000 lfm. Using a coefficient of rebound of 0.2, which is approximately the order of magnitude to be expected, the initial rebound velocity is 1000 lfm. Part of this velocity is dissipated by the jet and counter turbulence. Therefore, assume that the minimum control velocity is in the neighborhood of 500 lfm.

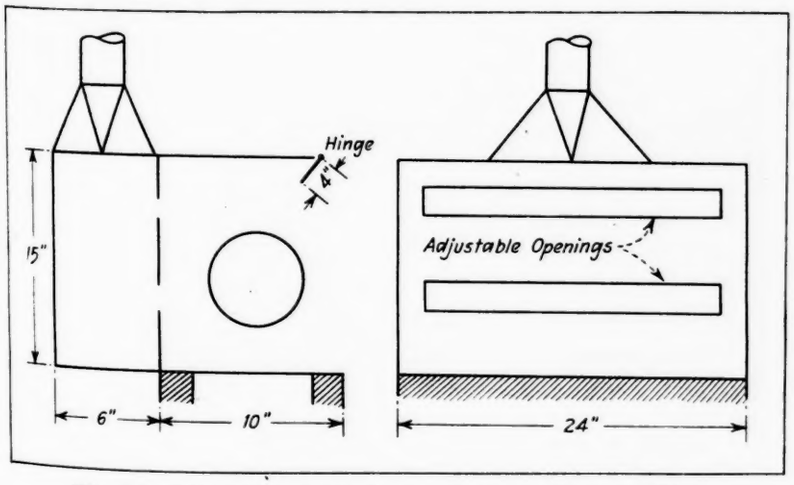


Fig. 87. Hood arrangement for metal spraying on lathe.

5. A possible hood design is shown in Fig. 87. This illustrates a rectangular enclosure with an opening 15 × 24 in. in area. It provides a 4 in. stiff hinged flange at the top edge. The hood rests on the lathe bench. At a distance of 10 in. from the opening (although this distance is optional) a partition with adjustable openings is provided. This allows for regulating the air flow so as to obtain a uniform distribution over the opening area, as well as areas of accelerated air speed near the work surface.
4. The control air volume will consist of that based on a controlling velocity of 500 lfm across the opening plus that volume that is required for removing the jet air plus the induced air, thus

$$Q = 500\ (24 \times 15/144) + 100 + 120 = 1470 \text{ cfm}$$

Example E. A continuous drying oven 30 ft long is steam-heated to a maximum temperature of 140F. Objects are heated within the enclosure, and certain explosive vapors are given off with density with respect to air of 3.2 at 70F. Air enters the inlet, into which the objects travel, and leaves from the opposite end. This air enters at 70F, and must be 140F at the point of exit. Controls are provided for this purpose. The total amount of liquid explosive solvent which must be removed during a drying cycle of 15 minutes is 990 cc and passes a specific gravity of 0.76. If the lower explosive limit of the vapor is 1.5% by volume, what volume of air must be provided so that the concentration will not exceed half this amount, or 0.7%?

1. The specific gravity of this special type of vapor at 140 F is

$$S.G. = 3.2\ \frac{460 + 70}{460 + 140} = 2.9$$

2. The volume of liquid vaporized per minute is $990/15 = 66$ cc per min
3. The weight of a cubic foot of air at 140F is

$$W = 0.075\ \frac{460 + 70}{460 + 140} = .067 \text{ lb per cu ft}$$

and the weight of 1 cu ft of vapor at this temperature is

$$W' = 0.067 \times 2.9 = 0.19 \text{ lb per cu ft}$$

4. The weight of the liquid evaporated during one minute is

$$66 \times 0.76 = 50 \text{ grams} = 0.11 \text{ lb}$$

5. The cubic feet of vapor produced per minute is $0.11/0.19 = 0.58$ cu ft at 140F
6. This volume of vapor must be diluted so that it is 0.7% of the air volume supplied at 140F. Hence $0.007 \times Q = 0.58$, and $Q = 83$ cu ft per min at 140F
7. At 70F, the volume of air required is

$$Q = 83\ \frac{460 + 70}{460 + 140} = 73 \text{ cu ft at 70F.}$$

These examples present the type calculations entering into the design of hoods to obtain control of an air contaminant. They are, of course, of a preliminary nature, but they suffice to show the magnitude of the control velocities and air volumes needed in certain practical problems. It must be remembered that few data are available on air flows necessary to control air-borne contaminants and these apply to operations which are of wide occurrence or are serious health problems. The previous articles of this series have covered information of this kind. The examples deal generally with types of conditions for which data are lacking, and give the necessary procedures to be followed.

(This concludes Mr. Dalla Valle's series on Exhaust Hoods)

Large Demand for Mineral Wool in 1942

According to information recently released by the United States Bureau of Mines, 41 companies operating 52 plants in 18 states turned out during 1942 a total of 513,072 tons of mineral wool products for insulation and other purposes valued at $31,656,927. A total of 5,624,106 ft of pipe covering was produced valued at $1,149,381.

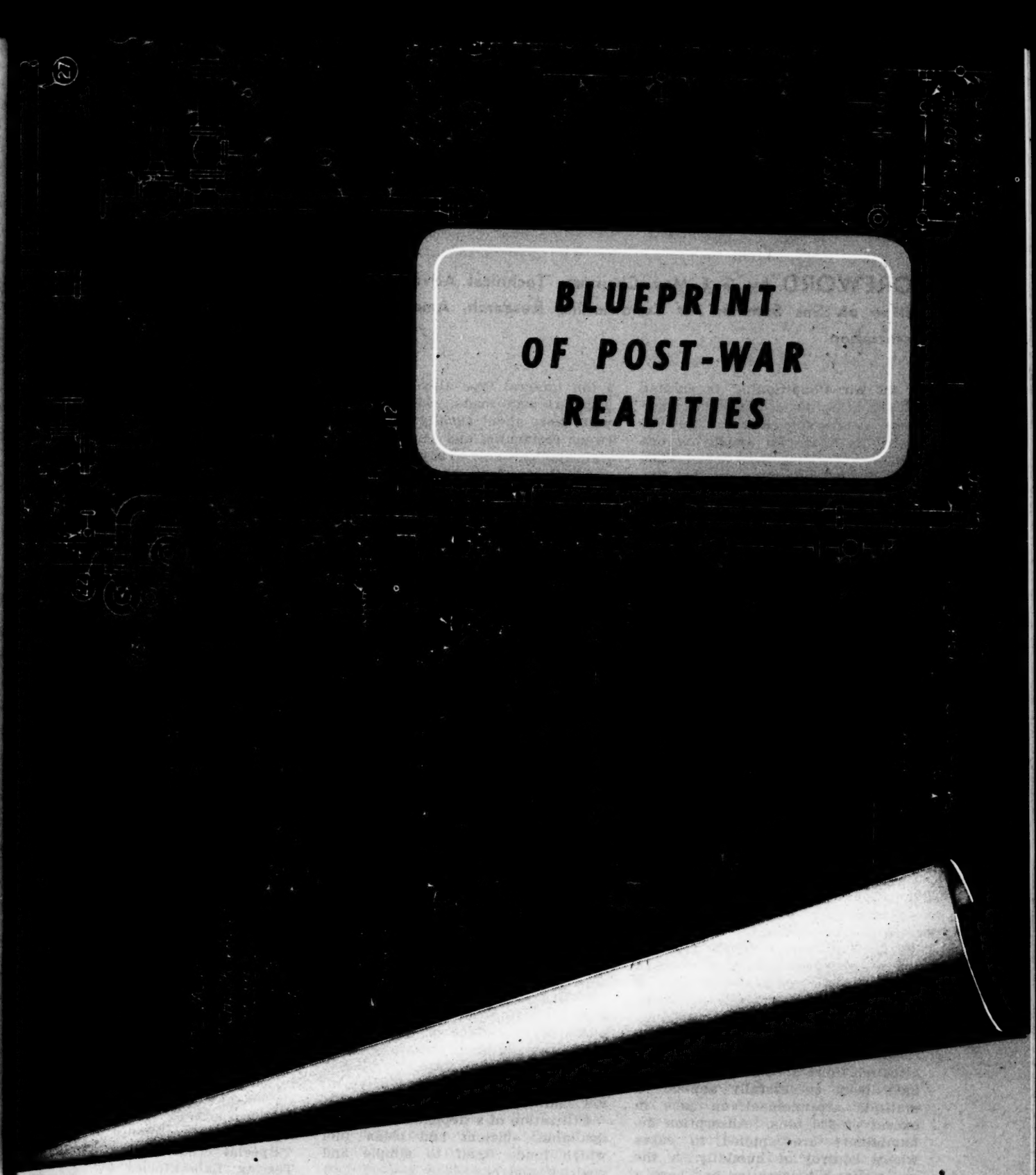
Of the pipe covering produced in 1942, $14,916,426 of the mineral wool was used in home insulation and $16,740,501 was used for industrial and military purposes.

Following is a summary of the production of mineral wool in the United States for the year 1942 both for home use and for industrial insulation:

PRODUCTION OF MINERAL WOOL IN THE UNITED STATES IN 1942

Product	Home Insulation		Industrial Insulation		Total	
	Short Tons	Value	Short Tons	Value	Short Tons	Value
Loose and granulated wool	241,693	7,729,321	11,523	416,396	253,216	8,145,717
Bats	86,716	4,960,812	60,346	4,624,546	147,062	9,585,358
Rolls	49,194	2,144,927	1,518	66,305	50,712	2,211,232
Felt, blocks, boards, mineral wool insulating cement, blankets, etc.	1,539	81,366	60,543	11,633,254	62,082	11,714,620
Pipe covering	—	—	5,624,106*	1,149,811	5,624,106*	1,149,881
Total	379,142	14,916,426	—	17,890,382	—	32,806,808

*Linear feet. Because of varying sizes of pipes, conversion to tons was not possible.



GAS for Summer Air Conditioning

Gas For Summer Air Conditioning

FOREWORD by G. E. May, Chairman, Technical Advisory Committee on Gas Summer Air Conditioning Research, American Gas Association.

GAS Air Conditioning is gaining the attention of architects, engineers, contractors and the gas industry as a system employing one fuel for providing year-round comfort. The same equipment that furnishes warm air during the winter months also supplies cool air during the summer, and controls humidity while heating or cooling—as well as ventilates and circulates clean air during all seasons of the year.

The present status of equipment design, installation and performance reflects the close cooperation between manufacturers, gas utilities, and American Gas Association committees concentrating on the development of high-quality gas year-round air conditioning. This work extends over the past decade, and the present advanced state of development will be evident to the reader after reviewing the following sections of this "Blueprint."

Air-conditioning systems operated by gas now exceed 100,000 tons of refrigeration capacity. These are made up of steam-turbine-driven centrifugal compressors, steam jet plants, absorption refrigeration systems and adsorption direct dehumidifiers. The first two types are generally applied to installations whose refrigeration requirements are in the range of 100 to 1200 tons or over. Absorption refrigeration systems have been used generally on installations varying from 3 to 40 tons, although there are cases where they have been successfully applied in multiple arrangement on jobs in excess of 300 tons. Adsorption dehumidifiers are applied to cases where control of humidity is the paramount consideration and have a vast market in the industrial field, though they are also used in connection with refrigeration equipment for expanding capacity and providing closer control of humidity; still another field covered in this "Blueprint," is direct application to comfort installations, in capacities ranging approximately from 3 to 15 tons.

The greatest growth in numbers of gas air conditioning installations occurred just prior to Pearl Harbor in the residential and small commercial field. Within a period of slightly less than two years, over three hundred installations of 3 and 5 ton integral type all-year air conditioners were made. Sales of these units were about equally divided between residential and small commercial applications. They were well received in areas where they were aggressively promoted and acceptance of them was rising rapidly.

Post-war planning on the part of manufacturers and many gas utilities is actively under way, following procedures in keeping with those recommended by the Committee for Economic Development. A great deal of publicity has recently been given through newspapers, magazines and radio, announcing that gas air conditioning is here and is proven. Regional meetings of manufacturers, utilities, architects, engineers, and contractors have served to initiate the formulating of definite plans for post-war action. A large market for such equipment is envisioned.

Listed below are features of gas air conditioning equipment which have received the favorable consideration of owners, operators, architects and engineers alike:

Use of the same prime energy source for both summer and winter seasons.

Ease of changing from heating to cooling and back again to heating, daily if necessary.

Quietness of operation, and freedom from vibration—features which allow the architect or engineer to locate the equipment in areas otherwise unsuitable.

Few moving parts, conducive to low maintenance cost and long life.

Utilization of a dependable, readily available, efficient and clean fuel which lends itself to simple and positive control.

In absorption refrigeration units, use of safe and inexpensive refrigerants (such as water), conforming to code requirements for air-conditioning system and operating at a vacuum, thus minimizing the loss of refrigerant.

Efficiencies which afford reasonable operating cost.

Along with these physical advantages, there are economic considerations. For example, summer air conditioning load normally occurs during the valley period of gas send-out, but in many cases is directly on the peak of the electric distribution systems. In the first case, there is surplus distribution capacity available, while in the latter, added investment in transmission (and in some cases generating) facilities is required. This may influence the future relative cost of gas and electric energy for air conditioning services. Yet, even at present, gas equipment may be operated at lower cost in many areas.

In the past, many building owners have felt justified in paying a premium for gas as fuel for winter heating on the basis of stated advantages. Costs frequently were higher than may be expected for the future, because they reflected a good deal of "learning costs" and inexperience in the field as well as expensive hand production at the factory. It is reasonable to believe that as organized field procedures with trained personnel take the place of inexperience, and mass production replaces handmade methods, that substantial reductions in investment costs will be effected.

THE sections of this "Blueprint" which follow describe the operating principles and give the results of American Gas Association tests of Bryant, Servel and Williams equipment. These tests were conducted with the thoroughness characteristic of a scientific laboratory and were originally presented in the American Gas Association Research Bulletin 18. The findings not only confirm manufacturers' data and field experience, but place valuable industry-wide approval on each type of equipment. These data, along with other data not included in this summary, constitute the first dependable information ever published on the design of gas air conditioning systems with respect to equipment requirements and performance.

Special credit is due the A.G.A. Testing Laboratories at Cleveland for the thoroughness with which the entire testing program was conducted, and for the informative analysis of the test results. The greater part of this work was done under direction of Dr. F. E. Vandaveer and W. B. Kirk, of the Laboratory staff to whom the industry expresses its appreciation.

The traditional and conventional method of obtaining cooling and dehumidification for summer air conditioning is by compression refrigeration or by so-called evaporative cooling, as an adjunct to, or independently of winter heating facil-

ities. The newer principles now being perfected under auspices of the gas industry recognize both heating and cooling as thermal processes capable of being efficiently accomplished by apparatus which is an integral component of a unified system. The initiative shown by pioneers in this enterprise is evidence of their determination to secure performance and user satisfaction superior to those attained by types of equipment that heretofore have preempted the field.

DESIGN OF UNITS

THE two main functions of summer air-conditioning systems are cooling and dehumidification of the air. Cooling effect in such systems is produced by refrigeration. Drying effect is obtained also by refrigeration, which condenses moisture out of the air simultaneously with the cooling, or by absorbing the moisture in some material which has high affinity for water vapor. (Drying, of course, is frequently a year-round feature.)

There are three basically different systems commercially available for producing these cooling and drying effects. The first is the compression refrigeration system well known to air conditioning engineers and which, by continuously compressing and expanding a fluid, creates a cold contact surface for both cooling and drying the air. A second system, known as the absorption refrigeration system, also effects cooling and drying through use of a cold surface. However, instead of using a compressor to circulate the refrigerant vapor, it uses a solvent having affinity for the refrigerant vapor to pick up the vapor in an absorber. This vessel corresponds to the suction side of the compressor. The solution of refrigerant and solvent is then pumped to another vessel known as the generator or heater, where heat drives the refrigerant vapor out of solution at the higher pressure. This corresponds to the discharge side of the compressor. The simplified principle of this method is shown in Fig. 3.

Both of the foregoing systems require energy for operation—electric or mechanical power for the compression system and heat for the absorption system.

The third system, known as the dehydration system, operates by bringing the air into direct contact with a special material, either solid or liquid, which has high affinity for moisture. This dehydrates the air, but in the process raises its temperature, necessitating removal of this heat by means of a dry-air surface cooler, to restore the original temperature. The air stream may be further cooled by evaporating water into it. In such cases, a surplus of drying action must be produced in the dehumidifier, so that when moisture is added to give the extra cooling effect, the air is also corrected for moisture. Heat is employed to dry the adsorbent material, enabling it to be used over and over again.

All three of the systems mentioned can be employed interchangeably with any source of heat or energy. For example, the compressor in the first system can be driven by a gas engine; an absorption system can use steam from any source or hot exhaust gases from an engine, and the dehydration system can be used with fuels other than gas, or with steam. In the material which follows, however, gas fuel as a source of energy is featured for two reasons: first, several of the principal units described are available primarily for gas and in one case for gas only; second, operating data have been made available for gas by the work of the gas industry and cooperating manufacturers.

In the pages which follow, especial attention is paid to the second and third types of system; equipments now or in the post-war period to be available are featured. The equipments are all post-war realities —not laboratory toys or dreams. All have been extensively tested in both laboratory and field.

Design of Servel Units

Servel, Inc., Evansville, Ind., has developed self-contained all-year air conditioners with two sizes of summer cooling sections. Heating sections of these air conditioners are available in sizes ranging from 60,000 to 180,000 Btu per hr gas input rating, and cooling sections are of 3 and 5 ton capacity with gas inputs ranging from 60,000 to 100,000 Btu per hr.

Cooling is produced in Servel units by the absorption refrigeration principle. Though the condenser pressure is maintained at a level above the evaporator pressure, this pressure elevation is slight and heat is used to circulate the solvent through the cycle, eliminating the necessity for a solution pump. This system is sealed and operates under a high vacuum at all times. Pressure within the condenser is approximately 1 lb per sq in. absolute while that within the evaporator is approximately 0.15 lb per sq in. Recent units are charged with lithium bromide as a solvent and with water as the refrigerant.

Fig. 7 illustrates the operation of this unit. The refrigeration generator consists of several small vertical tubes within a cylindrical shell.

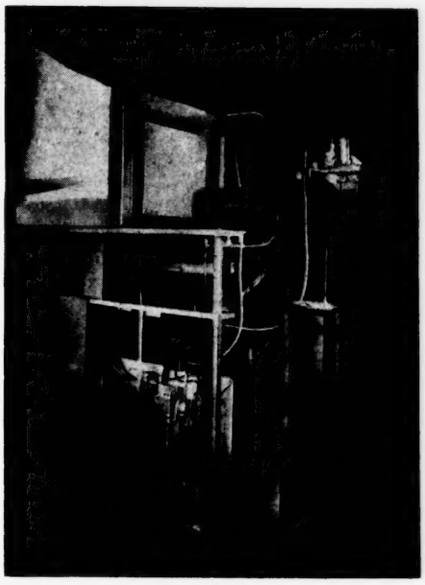


Fig. 1. View of side and back of a Servel 3-ton conditioner with jacket removed.

A solvent-refrigerant mixture partially fills these tubes. Steam is supplied to the outer shell of this generator from the steam boiler. Steam is condensed on the tubes containing the mixture, drops to the bottom of the outer shell, and flows to a condensate-return pump. As steam is applied to the generator, the refrigerant is evaporated from the mixture and the solvent (liquid) is raised by a thermal siphon or "percolator" action together with the refrigerant vapor to a vapor-separating chamber at the top of the generator. Here the solvent liquid is separated and flows to a liquid heat exchanger and thence to the absorber.

Refrigerant vapor passes from the separating chamber through a tube connected to a condenser where it is condensed by the cooling water flowing through condenser tubes. This cooling water is supplied from an external source and flows through the absorber coil and purge chamber, interconnecting pipe, condenser, and is then discharged from the unit to a sewer or to a cooling tower for re-use. Refrigerant liquid flows from the condenser through the evaporator inlet, a long U-tube. This tube contains refrigerant liquid (water) at

all times and forms a seal to maintain the pressure differential between condenser and absorber. Refrigerant liquid flows from the U-tube into a flash chamber where part of it vaporizes due to a lower absolute pressure which exists at this point. The small amount of refrigerant evaporated lowers the temperature of remaining refrigerant liquid. Refrigerant vapor formed in this chamber passes around a baffle and through a vapor outlet pipe, and

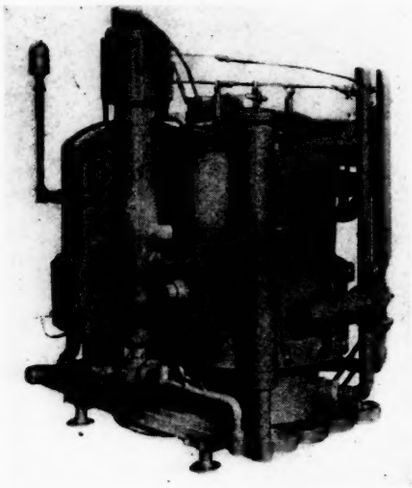


Fig. 2. Williams Air-O-Matic 7-ton absorption unit with heater-condenser section at left, pump in center, absorber-water chiller section at right, heat exchanger below.

is discharged into the absorber. Chilled refrigerant liquid from the flash chamber passes through an outlet tube at the bottom and flows to the evaporator or cooling coil.

Air to be conditioned (cooled and dehumidified) flows over the finned tubes of the cooling coil. The heat absorbed from air evaporates the refrigerant liquid within the coil. Evaporation of the refrigerant liquid at low temperature is possible because of the high vacuum in the cooling coil. Refrigerant vapor formed in the evaporator escapes to the vapor header and is then drawn downward into the absorber where solvent liquid absorbs it.

As explained, solvent liquid flows from the separating chamber above the generator, to a liquid heat exchanger. The solvent liquid cooled by this heat exchange flows to the absorber inlet pipe. A by-pass line is connected at this point to the water cooled purge chamber for the collection of non-condensible gases gathered from the absorber. The non-condensible gases are drawn through a siphon tube to a receiver tank equipped with a collector and purge valve. Gases are delivered to this receiver tank. A sight glass is located in the line leading to the receiving tank so that visual inspection under normal full-load operation will show whether or not the unit needs evacuating.

The absorber coil consists of a multiplicity of horizontal tubes. Solvent liquid flowing into this section is distributed over the entire surface of the coil to obtain the greatest possible area of contact between the solvent liquid and refrigerant vapor. The preheated solvent-refrigerant mixture leaving the liquid heat exchanger flows to the leveling chamber. The chamber is usually only partly filled with mixture which furnishes a reaction head for the generator system so that refrigerant vapors formed in the generator tubes can raise the solvent liquid up to the vapor-separating chamber. This leveling chamber supplies solvent-refrigerant mixture to the generator through an inlet pipe connected to the base of the generator. A connecting pipe between the leveling and vapor-separating chambers also serves to equalize pressure between these two sections.

When it is desired to heat rather than cool air with a Servel air conditioner, the operation of changing temperature dial setting and throwing a small finger switch on the room thermostat control serves to divert steam from the boiler through a pipe to a heating coil located in the air stream. This operation is accomplished automatically by a two-position steam diverter valve, activated by a small motor, which shuts off steam supply to the cooling unit and opens the steam passageway to this heating coil. Condensate from the heating coil returns to the boiler through a return tube.

As an optional feature to reduce water consumption, Servel offers a matched mechanical draft water cooling tower for each size of summer cooling section.

Design of Williams Units

Williams Oil-O-Matic Heating Corp., Bloomington, Ill., has developed three Air-O-Matic absorption units of chilled water type, suitable for large residences and small commercial applications with ratings of 7, 15, and (perhaps) 32 tons. This type of unit is designed to deliver chilled water at temperatures as low as 34F and requires a supply of low pressure steam and cooling water.

A two-pressure system is employed in this unit. Low-side pressure in the machine is ordinarily maintained between 21 and 24 in. mercury vacuum. Operating high-side pressure ranges between 1 and 3 lb per sq in. (gage). When the unit is not in operation these pressures equalize at 15 to 20 in. mercury vacuum, depending on room temperature. The unit is charged with methylene chloride as refrigerant and dimethyl ether of tetraethylene glycol as solvent. According to the manufacturer, both of these organic compounds are essentially non-toxic, non-irritant, non-corrosive and chemically stable.

Principal parts of this system consist of an absorber, heat exchanger, solution pump, heater, condenser, and water chiller, as shown in Fig. 8. Low-pressure steam (normally 10-12 lb per sq in. gage) is introduced to the steam space of the heater where its heat is transferred to solvent refrigerant mixture in a series of coils. Steam condensate is normally returned to the boiler which is not integral with the refrigeration unit.

Heated solvent-refrigerant mixture is discharged from heater coils into an inner vapor release chamber where refrigerant vapor separates from solvent liquid at a pressure of 1 to 3 lb per sq in. (gage), collects at top of this chamber, and flows down to the condenser, where refrigerant vapor is condensed by cooling water circulated through coils. Refrigerant liquid is discharged from the condenser through a drain pipe and is passed through the solvent liquid-return heat exchanger to the refrigerant float valve. The float valve, actuated by refrigerant liquid level in the water chiller, controls flow of refrigerant liquid through the feed line to the water chiller.

Refrigerant liquid in the chiller is vaporized at low pressure (21 to 24 in. mercury vacuum) by absorbing heat from water circulating to the air conditioner. Refrigerant vapor passes up through center of a flat baffle plate and is vented into top of absorber where it is absorbed by solvent liquid sprayed from a bank of nozzles over cooling coils through which cooling water is circulated prior to its passage to the condenser to remove the heat of absorption. The cool solvent-refrigerant mixture is drained from absorber into suction chamber of solution pump and is then pumped through outer passages of heat exchanger where it picks up heat from hot solvent liquid returning to the absorber from the heater. Solvent-refrigerant mixture is discharged from heat exchanger into heater coils. This completes the main cycle of refrigerant travel through the system.

Liquid refrigerant entering the water chiller contains a small amount of solvent liquid (usually about 0.15%). This solvent liquid must be removed from the chiller to prevent its accumulation as it inhibits evaporation. This is done by continuously bleeding off a little liquid through the orifice below the

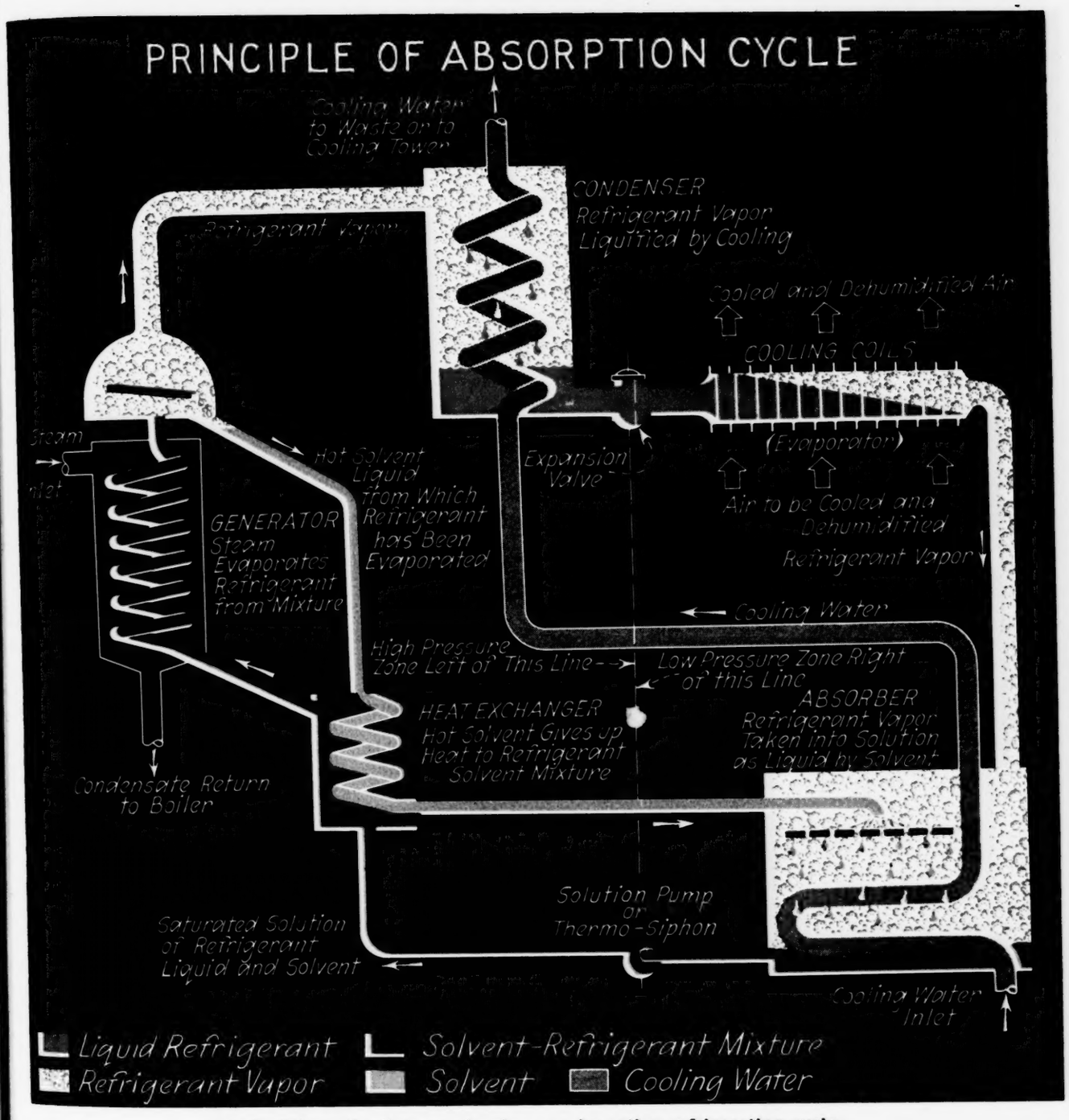


Fig. 3. Schematic diagram showing an absorption refrigerating cycle.

chiller. This mixture is then passed through the heat exchanger where the refrigerant fraction is evaporated by refrigerant liquid from the condenser and the solvent fraction is raised by thermosiphon action to the top of the absorber. Because the orifice is properly sized there is no heat loss, the vaporizing refrigerant serving to precool liquid refrigerant prior to its introduction to the chiller.

Cooling tower water is supplied through a feed-line to cooling coils in absorber and is discharged to a pressure control valve which, when unit is started up, by-passes flow of cooling water back to the tower, thus preventing flow through the condenser until normal refrigerant-vapor pressure is attained. When normal pressure is established it actuates the pressure-controlled valve which directs the water either through the condenser or through a by-pass line to the cooling tower or to waste. This action permits maximum cooling of absorber at all times with maintenance of proper condenser pressure to effect good refrigerant feed.

In addition to a manual cut-off switch, the following limit controls are supplied to shut off the solution pump under conditions indicated:

1. Low-limit steam control, if steam pressure is less than that necessary for efficient operation.
2. High-pressure cut-out, if condenser pressure exceeds 5 lb per sq in.
3. Low-temperature safety cut-out, if refrigerant liquid temperature in water chiller drops below 34F.
4. Overload safety cut-out to guard against excessive current; stopping solvent flow immediately stops refrigerating action of the system.

Design of Bryant Units

The Bryant Heater Co., Cleveland, Ohio, manufactures rotary-drum silica gel dehydrators which may be employed for commercial and resi-

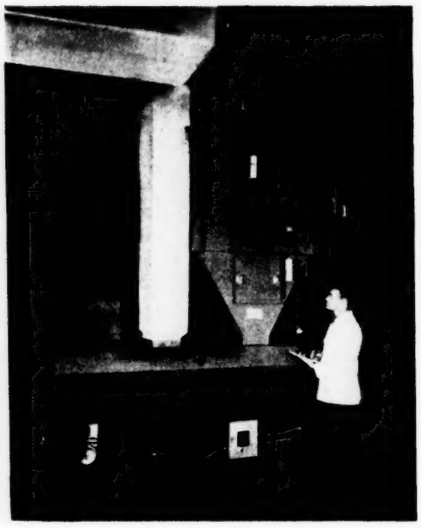


Fig. 4. A Bryant rotary type silica gel dehumidifier and accessory equipment, with jacket removed.

dential air conditioning as well as for industrial processes. These are available in four sizes with nominal air capacity ratings of 500, 800, 1,300, and 2,900 cfm. For comfort cooling purposes, companion dry air coolers (water coils for removing sensible heat from dried air) and resaturators (partial rehumidifiers to provide further air cooling) are obtainable from the manufacturer. The cycle is illustrated in Fig. 9.

Each of the three component parts of this system is employed to perform a separate and distinct air conditioning function. The dehumidifier (or dehydrator) removes moisture from circulated air. In so doing it converts latent heat to sensible heat in effluent air and also adds processing heat (including so-called "heat of wetting") in sensible form. The dry air is then passed through a surface cooler where its temperature is reduced. Air temperature may be further reduced in the resaturator where some sensible heat is converted back to latent heat by evaporation of water into the air. This latter element makes it possible to reduce the dry bulb temperature of delivered air below the temperature of available cooling water.

Referring to Fig. 9, air to be conditioned is drawn into the dehumidifier casing through inlet air opening and passes through walls of a hollow revolving drum constructed of two concentric cylinders of fine mesh screen with the silica gel in granular form confined between them. This silica gel adsorbs moisture from air passed through it. Interior of the hollow drum, as well as surrounding space, is divided into two comparatively airtight sections by means of wiper blades attached to a dividing partition which separates the air drying chamber at right from reactivation chamber at left but permits rotation of the drum through both chambers. Dried air at high temperature is drawn from the interior of the right half of drum through an outlet opening by a centrifugal blower and is passed through cooling coils in the dry air cooler where its temperature is reduced to a point approaching the inlet water temperature before it enters the resaturation cooler.

In the resaturation cooler, air is forced through a rotating drum made of large mesh screen. To insure passage of air through this wire screening, the drum is sealed at its top by a rubber wiper blade and at its bottom by sump water in which the drum is mounted. Sump water is picked up by the revolving screen and some is vaporized in the air stream, thus causing a drop in dry bulb temperature.

The dehumidifier drum is rotated at a rate of approximately one revolution in seven minutes so that silica gel which has become saturated with water is transported through the dividing partition and into the reactivation chamber where it is dried and prepared for further use. In accomplishing this, air is drawn from outside the unit through the reactivation air intake. Most of this air passes up and around a heating element which contains gas burners, mixes with products of combustion from these burners, and is then drawn inward through the upper left portion of the drum where its heat drives water from the silica gel. Hot moist gases flow from interior part of the drum through an opening in front and are discharged by an exhauster through a stack to the outside. After silica gel has been dried it is quite hot. To cool this substance so that it will adsorb moisture when it reaches the air drying side of the unit, some unheated intake air is drawn through a compartment below the heating element and thence through the drum wall by the activation air fan. This cooling compartment is below the heating element and is also made relatively air-tight at the drum surface by means of a metal wiper blade.

Effects of these three processes on circulating air are exemplified by the following data obtained during a standard cooling capacity test. Air entering the system at 80F dry bulb, 67F wet bulb contains 78 grains of moisture per lb of dry air, 19.2 Btu per lb of air sensible heat, and 12.3 Btu per lb as latent heat, a total of 31.5 Btu per lb. In passing through the dehumidifier, total heat content is increased from 31.5 to 36.6 Btu per lb due to heat of adsorption (heat of wetting), friction to air flow, heat transfer from reactivation chamber partition, etc. While sensible heat content is raised to 30.8 Btu per lb, latent heat content is decreased to 5.8 Btu per lb. Air leaves the dehumidifier at 128F dry bulb, 73F wet bulb, and has had its moisture content lowered from 78 to 35 grains per lb. It is then forced through the dry air cooler where 10.2 Btu per lb of sensible heat is extracted by cooling water flowing through coils so that outlet air contains a total of 26.4 Btu per lb, comprised of 20.6 Btu as sensible heat and 5.8 Btu as latent heat. Outlet air temperatures have been reduced to 86F dry bulb, 60F wet bulb,

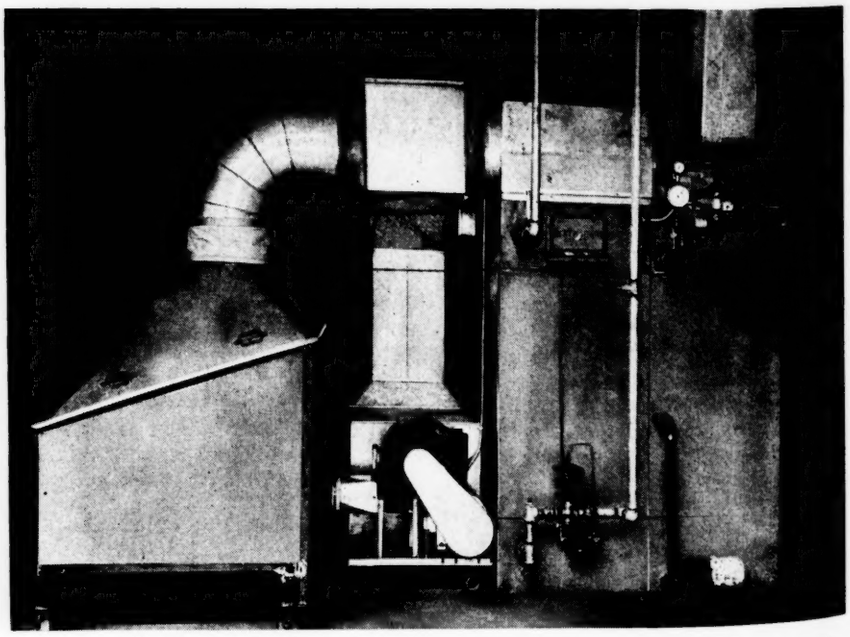


Fig. 5. A Surface Combustion Kathabar package-type dehumidifier in an industrial process installation.

while moisture content is constant at 35 grains per lb. To reduce dry bulb temperature (and sensible heat content) still further, water is evaporated into this air in the resaturator, so that some sensible heat is converted to latent heat without changing the total heat content appreciably. Air leaves the resaturator at 67F dry bulb, and 60F wet bulb and contains 73 grains of moisture per lb, 11.5 Btu per lb of latent heat, and 18.5 Btu per lb of sensible heat.

In addition to the units described, Bryant makes two sizes of stationary bed type dehumidifiers which have air capacities of 2,700 and 5,000 cfm respectively. These units are divided into two sections so that one section may be reactivated while the other section is utilized for drying air. Automatic dampers control these operations. Gas input ratings are 262,000 and 525,000 Btu per hr for the smaller and larger units, respectively, while electric power used is 3 and 5 kw.

Although experimental work conducted in the A.G.A. Laboratory was confined to the units mentioned in the foregoing, plus the Mills unit mentioned later, other gas-using air conditioning units are available for larger commercial and industrial installations. These include:

Design of Kathabar Units

Surface Combustion, Toledo, Ohio, has developed the Kathabar system for dehumidifying (or humidifying) and cleaning air. This system employs a contactor cell in which circulating air is passed through a spray of Kathene (lithium chloride brine). If excessive moisture is contained in air it is absorbed in this brine, which then flows to a regenerator where the solution is regenerated by heating it and passing outside air over it to carry off the excess water vapor. In regenerating, Kathene brine is circulated through tubes in a steam heater, sprayed back into the regenerator, and pumped through a tubular cooler heat exchanger before being returned to the contactor cell. Dried air from the contactor cell may be either heated or cooled by means of water or steam coils. A flow chart of this process is shown in Fig. 13. If insufficient moisture is present in air, the system may be reversed. In other words, moisture would be given off by the Kathene and picked up by circulating air. Whether moisture is added to or removed from circulated air depends on the setting of an automatic humidistat control which controls temperature (and vapor pressure) of the brine. Scrubbing action of Kathene spray, according to the manufacturer, deodorizes the circulating air.

Fig. 6. Two Pittsburgh Lectrodryers of the type used for industrial dehumidifying.

A series of four standard self-contained units, including contactor and regenerator sections, and with capacities of 750, 1,500, 3,500, and 5,000 cfm, are available. For built-up systems contactor cells are made in two sizes, the smaller having an air capacity of 3,000 cfm and the larger 4,000 cfm. For large installations these cells are combined in batteries. Regenerating systems are sized as required to handle these batteries.

Design of Lectrodryer Units

Pittsburgh Lectrodryer Corp., Pittsburgh, Pa., specializes primarily in air and gas drying equipment for all sorts of industrial processes. Air to be dehumidified and heated reactivation air pass alternately through beds of activated alumina serving as adsorbent. As with silica gel, dehydration is accompanied by sharp rise in dried air temperature. If cooling is desired in addition to dehumidification, water coils are supplied. These are either embedded in the alumina beds or may be installed in outlet air ducts. Standard units are produced in sizes ranging from 350 to 15,000 cfm air capacity, and can be used with gas, steam, or other source of energy. The arrangement of one type Lectrodryer is shown in Fig. 15.

Other Methods Using Gas

There are a number of other variations of equipment arrangements and cycles which can, if desired, be used to operate on gas.

One of them is the Mills (Mills Industries, Inc., Chicago) unit, at present in the development stage. This appliance is of the gas-fired absorption type for domestic and small commercial applications. A few experimental models with a nominal rating of 3 tons refrigerating capacity have already been made and subjected to field tests. Although production models have not been made available one of these units was examined and tested at the AGA Laboratories. It operated on a gravity absorption principle with ammonia refrigerant and water solvent. The manufacturer's development engineers are now modifying the design and for this reason no test results are reported here.

In this unit, generator and analyzer tank is fitted with heater coils which extend over gas burners in a manner resembling a conventional side-arm water heater application. Ammonia (refrigerant vapor) is driven from solution at a pressure of about 185 lb per sq in. and a temperature of approximately 260F. This hot refrigerant vapor is slightly cooled in a rectifier where the small amount of water (solvent liquid) evaporated from the mixture is condensed and returned to the analyzer. Refrigerant vapor is then further cooled and liquefied in a conventional evaporative type surface condenser and flows into a receiving chamber, still under generator pressure. From the receiving chamber it passes through an expansion valve into an evaporator coil where it is vaporized at low pressure (15 to 35 lb per sq in.) in the process of absorbing heat from the refrigerated medium. Refrigerant vapor passes from the evaporator through a check valve into the absorber where it is absorbed and taken into solution by solvent liquid from the generator, which is admitted through a fixed orifice. Solvent-refrigerant mixture from the absorber is passed through another check valve into a transfer chamber which is alternately subjected to high and low pressure. When this transfer chamber is filled

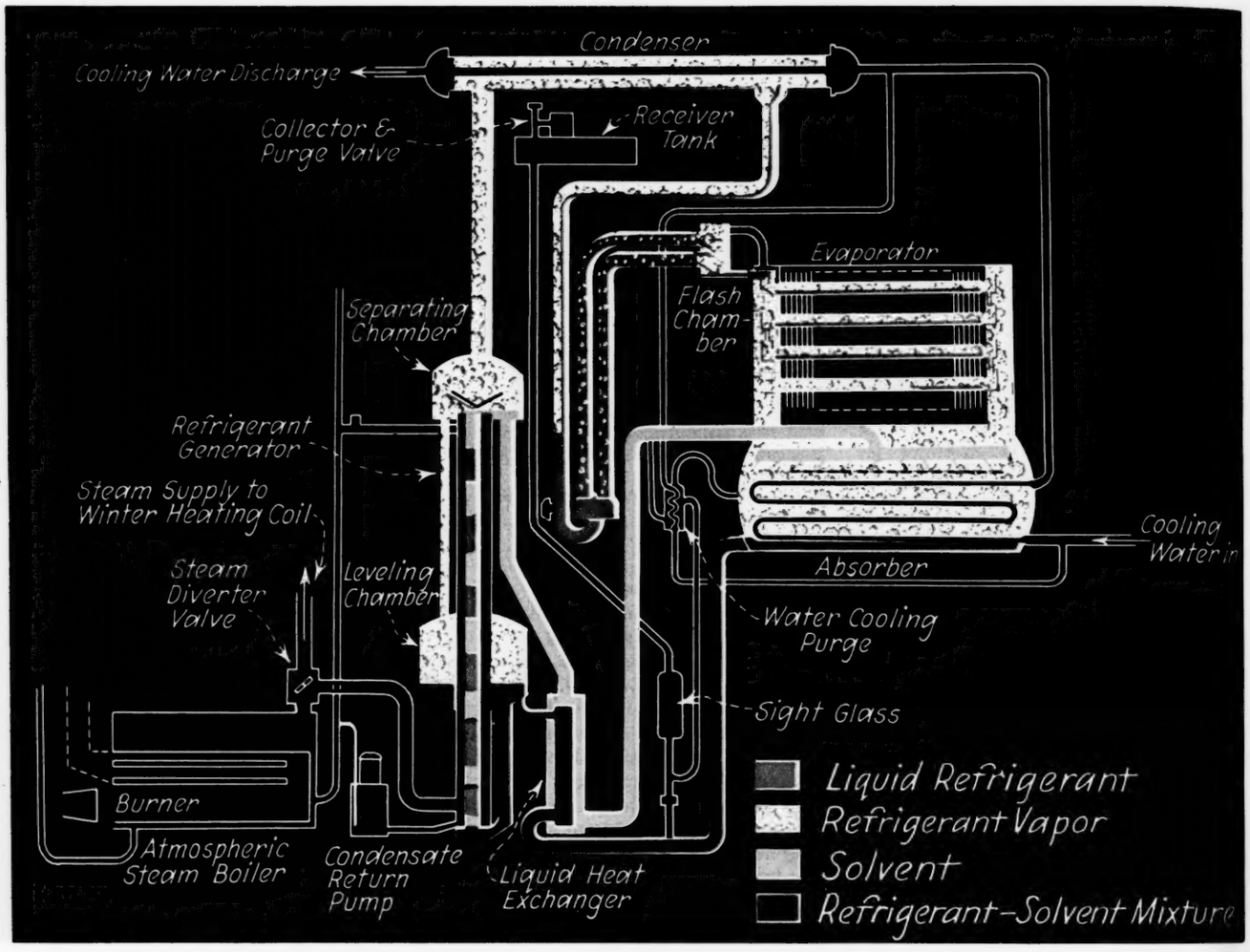


Fig. 7. Diagram of essential parts of the Servel all year conditioner.

with solvent-refrigerant mixture a balance valve admits vapors from the generator raising the pressure on the transfer chamber to equal that of the generator. The refrigerant-solvent mixture then flows by gravity from the transfer chamber through another check valve and back to the generator, passing through a heat exchanger on the way.

Carrier Corporation, Worthington Pump & Machinery Corp., Ingersoll-Rand Co., and York Corporation all have been concerned with turbine-driven compressor systems. Such systems could obviously utilize gas as the source of energy for steam generation for the turbines. In addition, both York and Worthington have a long background of experience with heat-operated ammonia absorption machines.

Finally, a compressor-type system can be powered by a natural gas engine, and installations of this type have been made by Waukesha Motor Co., and others. This type of installation has particularly interesting possibilities where the waste heat can be used for service water or process steam.

PERFORMANCE

THE following performance data for Servel, Williams and Bryant units were determined from tests conducted by the A.G.A. Laboratory.

As has been explained, the action of these various devices in performing their air conditioning function differs widely. To illustrate this difference in function the condition of the air entering and leaving the unit for each of the four devices tested has been plotted upon a section of the Bulkeley psychrometric chart (Fig. 12). In the case of the Bryant System, intermediate conditions between each element of the system have been plotted so that the true path taken in arriving at the final

TABLE 1.—TEST DATA OBTAINED ON SERVEL AIR CONDITIONERS UNDER STANDARD A.S.R.E. CONDITIONS

Data Recorded	Servel 3-Ton (No Cooling Tower)	Servel 5-Ton (No Cooling Tower)	Servel 5-Ton (Cooling Tower)
Outlet test air, dry bulb, F	62.1	60.3	60.7
Outlet test air, wet bulb, F	58.7	57.5	58.3
Flow rate of air, cfm	1219	2060	2030
Ccoling water flow rate, gpm	6.16	15.42	15.38
Heat input, Btu per hr	60,400	101,200	100,600
Electrical input, watts	607	840	1700
Latent heat extracted, Btu per hr	8970	13,450	12,180
Latent heat extracted, tons	0.75	1.12	1.01
Sensible heat extracted, Btu per hr	24,080	48,550	44,720
Sensible heat extracted, tons	2.01	4.05	3.73
Total heat extracted, Btu per hr	33,050	62,000	56,900
Total heat extracted, tons	2.76	5.17	4.74

condition is fully illustrated. This clearly sets forth the difference between the action of cooling-dehumidification and dehydration systems.

The entering condition for each of the plotted tests was approximately 80F dry bulb and 67F wet bulb which has been established by the A.S.R.E. as the standard entering condition for rating self-contained air conditioning units. Leaving conditions are as found by test.

Performance of Servel Units

First tests conducted by A.G.A. on the Servel air conditioners were made under conditions specified by A.S.R.E. as standard rating tests for equipment of this kind. This served the purpose of verifying the manufacturer's nominal rating and provided a basis of comparison for subsequent tests when other variations duplicating extremes of service conditions were applied. The 3-ton Servel system was tested for cooling capacity using city water for condensing, whereas the 5-ton unit was tested both with city water (no cooling tower) and with cooling tower. Test data are presented in Table 1.

The data show that a difference of 8.2% (0.43 ton) exists between total heat extracted by the 5-ton unit when operated with city water and with cooling tower. This is because the water cooling tower is not 100% efficient, i.e., circulating water cannot be cooled to 75F (wet bulb temperature of tower intake air). Thus, instead of 75F cooling water, the unit was supplied with water at a temperature approximately 8.5F higher during the cooling tower test.

Varying Inlet Air Conditions

By varying inlet air wet bulb temperatures over normal ranges which might be encountered in service, the ability of the 3- and 5-ton conditioners to extract heat was obtained by test. Results within the usual range of wet bulb temperature variation are graphically illustrated in Fig. 10. Since wet bulb temperature is a measure of total heat content and moisture content (with constant dry bulb temperature) of inlet test air, data on these factors are also plotted in Fig. 10. It seems evident from the shape of the curve, for total heat extracted (Fig. 13) that optimum performance of the 3-ton unit may be expected at a wet-bulb temperature of approximately 75F. This corresponds to a heat content of 39 Btu per lb and a moisture content of 122 grains per lb. The 5-ton unit also showed optimum performance at 75F wet bulb.

As inlet air wet bulb temperature is increased above 75F, trends shown by dotted line portions of curves in Fig. 10 were obtained. These dotted sections of curves are not to be extended further, although data obtained at higher wet bulb temperature indicated these general trends. In other words, on the Servel units employed in these tests, as inlet air wet bulb temperatures are increased above 75F rates of total and latent heat extraction begin to decrease rapidly. At the same time rate of sensible heat extraction decreases appreciably. It is recommended that performance data be secured from

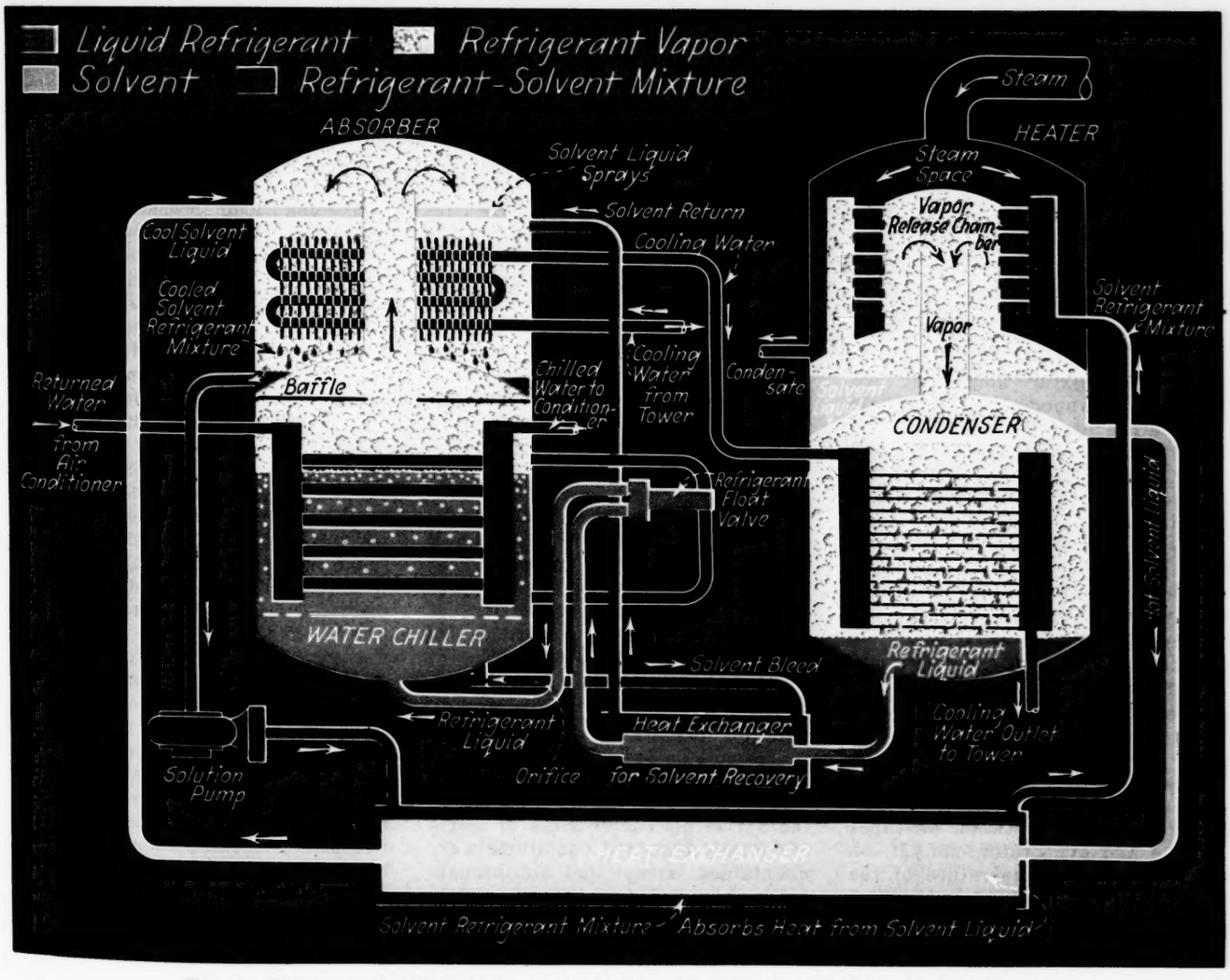


Fig. 8. Diagram of refrigerating cycle of the Williams Air-O-Matic absorption unit.

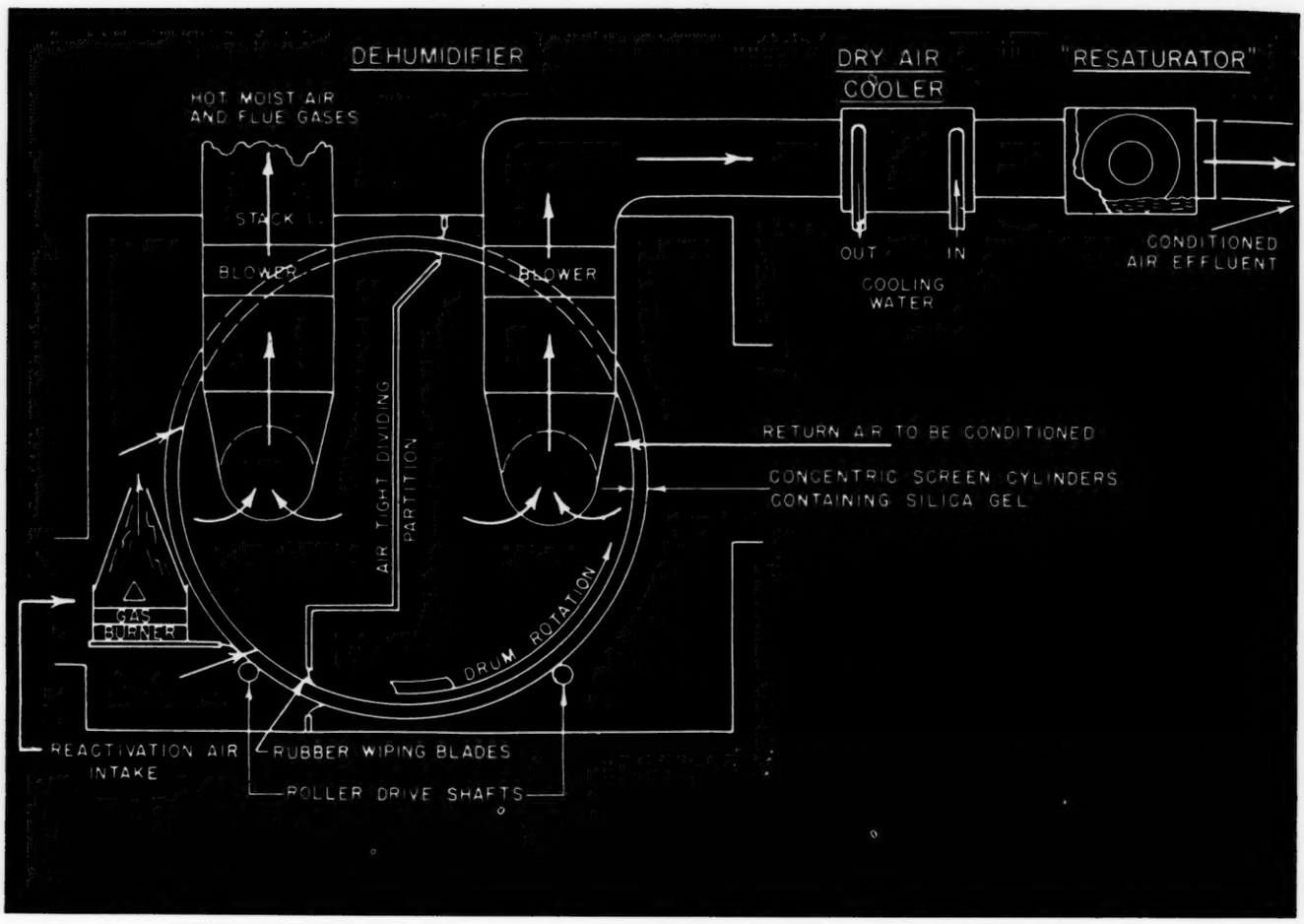


Fig. 9. Flow diagram of Bryant dehumidifier with dry air cooler and resaturator.

the manufacturer if operating conditions should involve wet bulb temperatures higher than 78F.

In studying the characteristics of these curves it should be remembered that heat content of air consists of sensible heat of the air-water vapor mixture and latent heat of the water vapor. If the dry bulb temperature of the mixture is relatively high at low total heat values, water vapor content and dewpoint will necessarily be low. Under these conditions, little or no condensation can occur, although a good temperature differential will exist between evaporator and air mixture. Whatever heat transfer occurs must be largely, if not entirely, a reduction in sensible heat of air-vapor mixture. This is usually a less efficient type of heat exchange than one wherein both sensible and latent heat contents of air mixtures are reduced, because this latter type of heat transfer serves to retain a higher temperature differential between air-vapor mixture and evaporator coil. If, on the other hand, temperature of the mixture is low, there is a smaller temperature differential, thereby reducing heat transfer. As total heat of air mixture is increased, either temperature differential or moisture content will correspondingly increase, with subsequent gain in heat absorbed by the refrigeration unit.

Effect of varying dry bulb temperature of inlet air, while maintaining reasonably constant wet bulb temperature at approximately 67.5F, is illustrated by curves in Fig. 11. It might be expected that the curve for total heat extracted should be a straight horizontal line because flow rate and total heat content of inlet test air were maintained practically constant. However, the slight decrease in total heat extracted at 95F dry bulb temperature indicates that heat exchange is somewhat less efficient when the proportion of sensible-to-latent-heat content is relatively high.

Change in Rate of Air Flow

A series of tests was conducted on the Servel air conditioners in which essentially standard conditions were maintained, except that air throughcut was varied from approximately 25% below to 25% above that employed in standard rating tests. Results of these tests reveal some interesting information on effect of increasing load without changing psychrometric conditions of inlet air.

Tests conducted on the 3-ton unit when air flow was varied 25% above and 25% below normal rate of 1,200 cfm showed that total sensible heat extracted increased with increase in air flow, whereas latent heat extracted decreased slightly. These results were to be expected, since with small air volume outlet air temperature decreases and a greater proportion of moisture in the air should condense out. Likewise, increased rate of air flow within a limited range might result in improved heat transfer and greater total heat extracted because of reduction in air film thickness around the chilled water coils and therefore reducing the resistance to heat transfer. These trends are indicated by curves in upper portion of Fig. 14. Although cooling capacity of this unit may be increased some 6% (from 33,000 to 35,000 Btu per hr) by increasing air flow 25%, dehumidification effect would be decreased about 11%, from 9,000 to 8,000 Btu per hr. An economic balance must accordingly be selected between these two characteristics. In territories with high humidities it might be preferable to operate at a lower air throughout

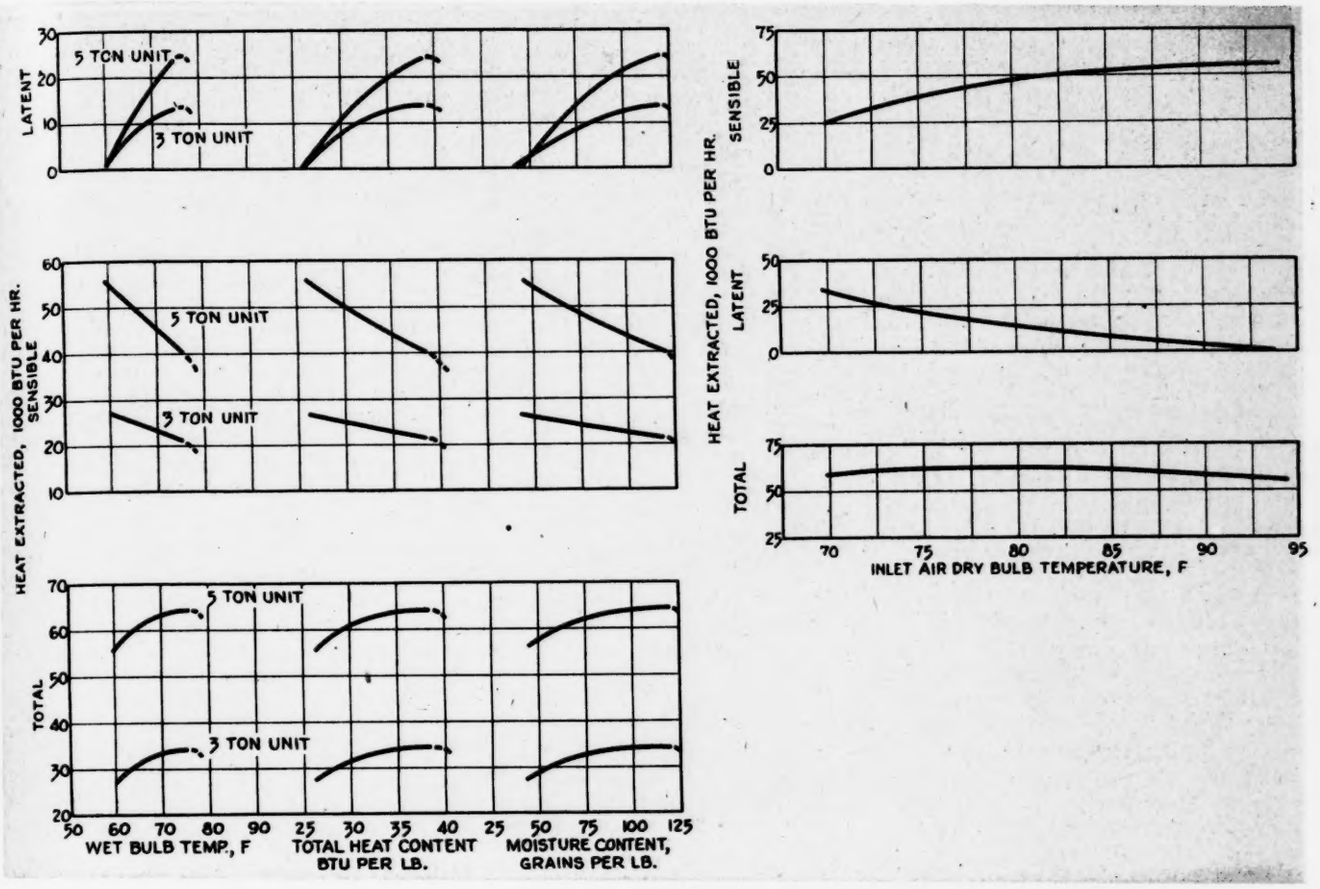


Fig. 10. (left) Effect of inlet air wet bulb temperature, total heat content, and moisture content on performance of Servel 3- and 5-ton all-year air conditioners. Dotted lines indicate trends and are not to be extended.
Fig. 11. (right) Effect of inlet air dry bulb temperatures on performance of Servel 5-ton all-year air conditioner.

and vice versa. It is also recognized that limitations are placed on air velocities attainable through the unit by static pressure through it, selection of blower as to capacity, motor, size, etc.

Data on the 5-ton unit, showing effect of variation in air flow from 1,500 to 2,500 cfm (2,000 cfm is normal), are also shown in Fig. 14. Up to a flow rate of approximately 2,200 cfm these curves follow trends similar to those shown for the 3-ton unit, but at higher flow rates total heat extracted gradually decreased, sensible heat extracted decreased appreciably, and latent heat extracted increased slightly.

To obtain an air flow rate of 2,500 cfm, it was necessary to replace the ½ hp motor for the blower with a ¾ hp motor. Furthermore, tests conducted in the manufacturer's laboratories had indicated that when air flows greater than 2,300 cfm were employed, refrigerant water passing through evaporator coils was not completely evaporated and cooling capacity would drop off. Curves in Fig. 14 verify these results and indicate that manufacturer's recommended air flow rate on this design of unit should be closely adhered to if economical performance is to be realized.

Cooling Water Variations

Cooling water temperatures are essentially dependent upon either temperature of available water supply or, in the case of tower equipped installations, upon climatic conditions. Except for well water or surface water fed by underground springs, even the temperature of available surface water supply is dependent upon wet bulb temperature of the atmosphere and solar radiation. Cooling water temperature, magnitude of supply, and cost of water are important factors in determining both rate of flow to be employed and whether a water cooling tower should be installed. Consequently, trends indicated in this

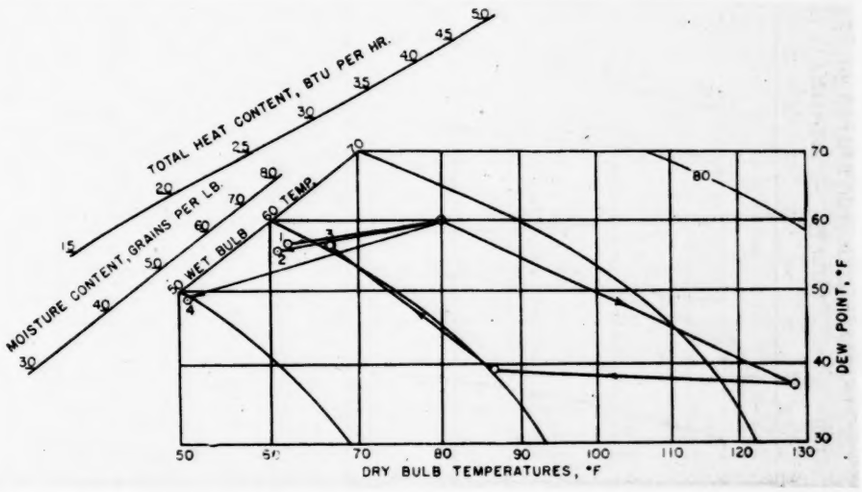


Fig. 12. Cooling and dehumidifying cycle for four types of air conditioners as represented on the Bulkeley psychrometric chart. (1) Servel 3-ton unit, 1200 cfm; (2) Servel 5-ton unit, 2000 cfm; (3) Bryant No. 14-R assembly, 1300 cfm; (4) Williams 7-ton unit, 1550 cfm.

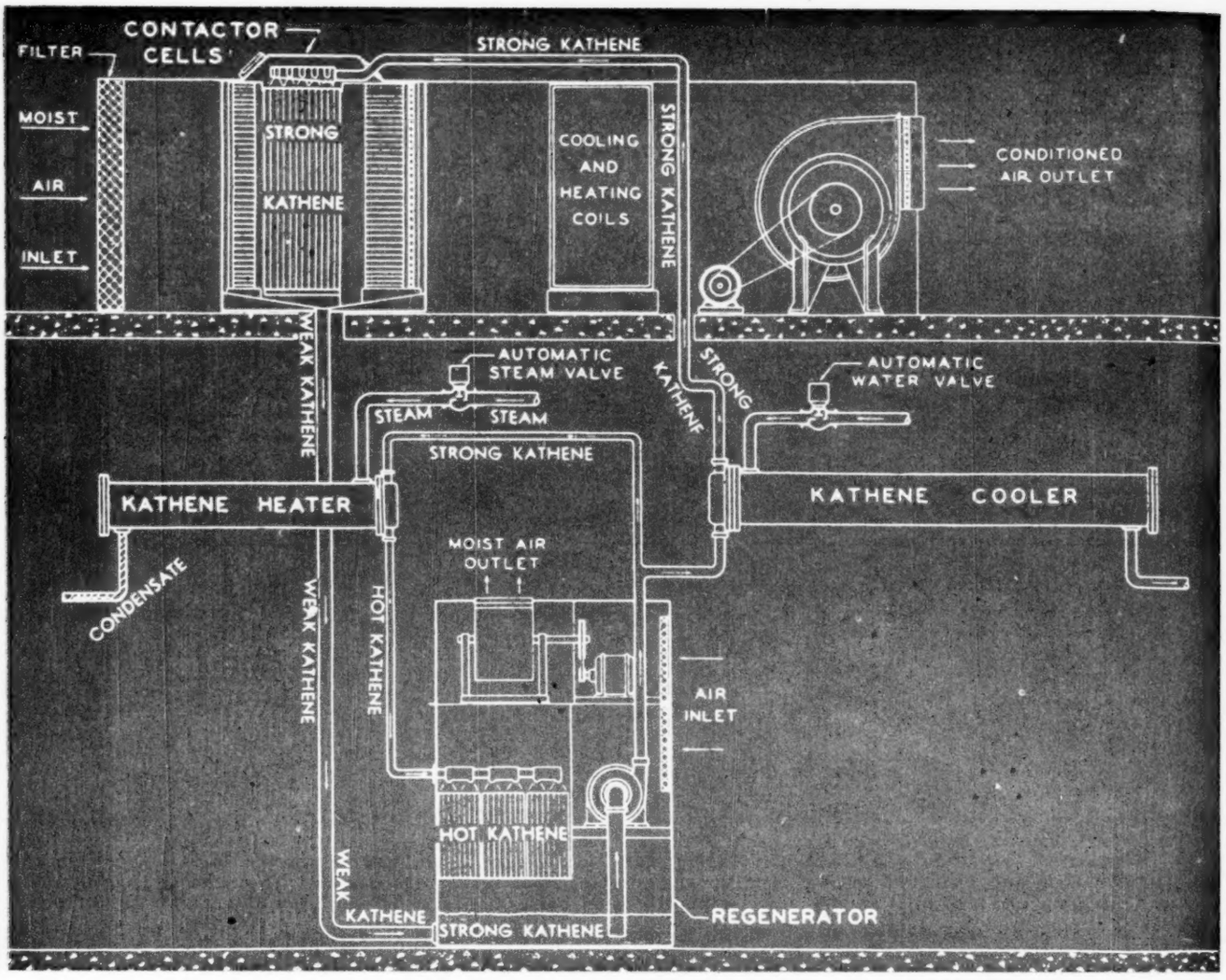


Fig. 13. Surface Combustion's Kathabar cycle for summer operation.

section are an index of performance that may be expected under various cooling water temperature and flow rate conditions.

Determinations of effects of cooling water flow rate and temperature on performance of the Servel 3-ton unit were combined, since this unit was equipped with a cooling water regulator and by-pass assembly for application where city water is used. This regulator shuts off the supply of cooling water automatically during "off cycles" of the refrigeration unit and regulates water flow during

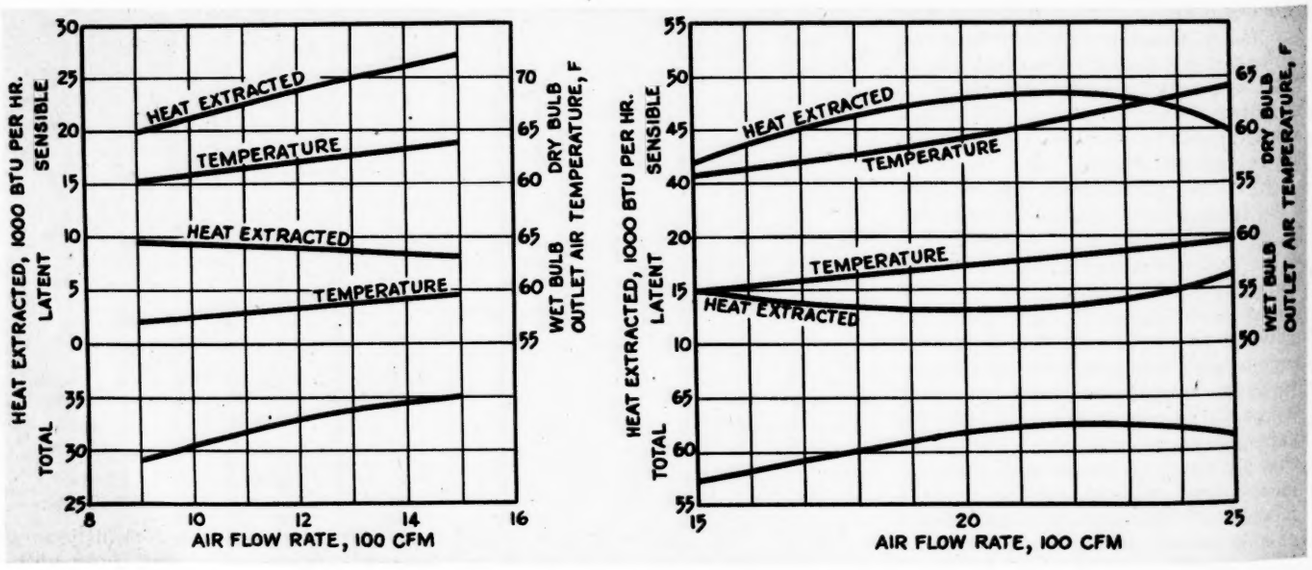


Fig. 14. Effect of change in air flow rate through Servel 3-ton unit (left), and on 5-ton unit (right).

"on cycles" so that a constant discharge temperature (approximately 100F) is obtained. For this reason any change in inlet temperature of cooling water automatically results in a simultaneous change in flow rate.

Results of tests conducted with cooling water inlet temperature varying from approximately 60 to 90F are presented in Fig. 16.

Increasing inlet water temperature to 88.8F served to increase water flow to as high as 8.1 gpm, although optimum conditions were apparently obtained with an inlet water temperature of 73.0F and a water flow rate of 6.2 gpm.

The 5-ton unit was supplied with a forced-draft water cooling tower, so that under full-load conditions its cooling water was constantly recirculated at an average rate of 15.4 gpm. No water regulator and bypass assembly is supplied with this type of system, but the tower water pump is connected in series with the air conditioner and they consequently "cycle on" and "cycle off" together. Effects of cooling water inlet temperature on performance of this air conditioner are shown in Fig. 16. Dissimilarity exists between these curves and those shown for the 3-ton unit because the 3-ton unit was tested with concurrent changes in both water temperature and flow rate, whereas the 5-ton unit was subjected to changes in water temperature with a constant flow rate of 15.4 gpm except in one test where a reduced flow rate of 7.5 gpm was employed with a cooling water inlet temperature of 60.7F. At this low temperature it was necessary to reduce flow rate to this figure to prevent shut down by the low temperature limit control on the flash chamber which protects the unit against temperatures which may freeze the solution. For this reason results of this particular test are connected with those of other tests by means of dotted lines shown in Fig. 16. Differential between inlet and outlet water temperatures was 36.4F in this test and averaged 18.4F in other tests.

Water Cooling Tower

In conjunction with tests on the 5-ton conditioner, observations were also made of performance characteristics of the Servel model water cooling tower under different conditions of intake air moisture and heat content, and air flow rate.

Effect of variation in intake air wet bulb temperature (or heat content) on performance is indicated in Fig. 17. Dotted lines indicate little change in total heat extracted, make-up water consumption, or bleed water flow rate as intake air wet bulb temperature was increased from 65.1 to 75.0F. Heat extracted from cooling water varied from 141,000 Btu per hr with a 65.1F wet bulb intake air temperature to 143,000 Btu per hr with a 75F wet bulb air temperature. Make-up water consumption was constant at 221 lb per hr and bleed water flow rate averaged 85.8 lb per hr. Inlet and outlet tower water temperature increased with intake air wet bulb temperature. As shown in Fig. 17, outlet water temperature was increased from 76.8 to 83.4F, and inlet water temperatures followed a similar trend.

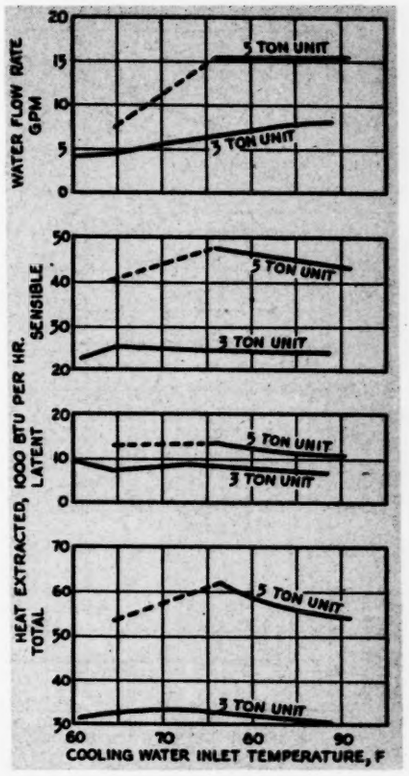


Fig. 16. Effect of cooling water inlet temperatures on performance of Servel 3- and 5-ton units. Note that the 5-ton unit was tested with a cooling tower, the 3-ton unit with city supply.

Essentially identical trends were obtained with air flow rates stepped up to 2,700 cfm. In one test, however, an unusually large quantity of water amounting to 118 lb per hr was bled from the tower and increased make-up water consumption correspondingly. It seems evident that if bleed water flow rate had been maintained reasonably constant at approximately 85 lb per hr, all corresponding lines drawn in Fig. 17 for air flow rates of 2,350 and 2,700 cfm would very nearly coincide. Furthermore, it may be generally concluded that intake air wet bulb temperature (and heat content) within a range of 65 to 85F has no appreciable effect on total heat extracted or make-up water required, providing bleed water flow rate is constant.

Varying air flow rate through the water cooling tower from 2,000 to 2,700 cfm increased electrical energy consumed by the tower motors from 815 to 1,040 watts. This is illustrated in Fig. 18. Note that almost a constant differential was maintained between inlet and outlet water.

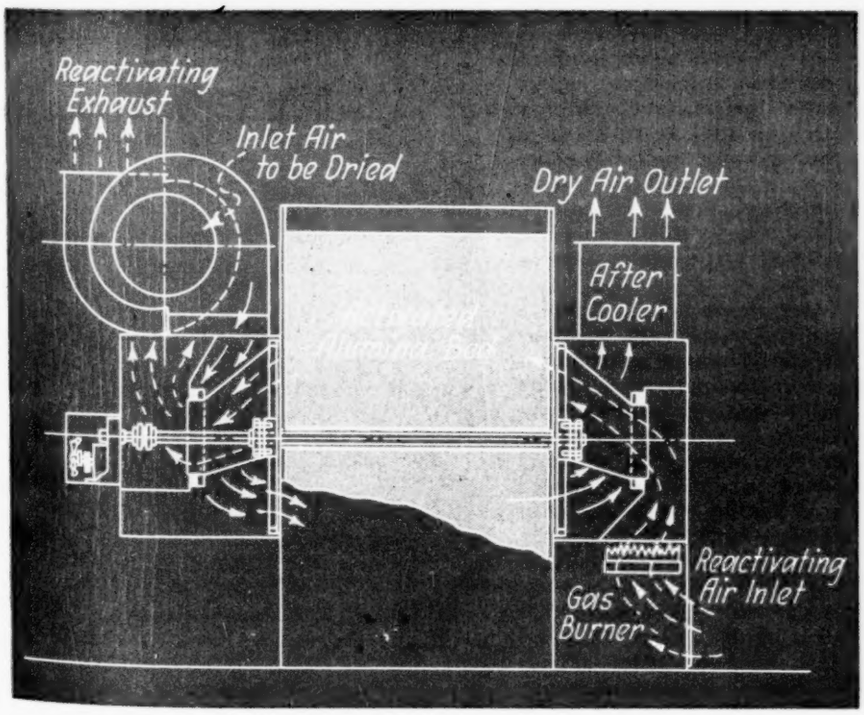


Fig. 15. Schematic diagram showing arrangement of one type of Pittsburgh Lectrodryer using activated alumina as drying agent.

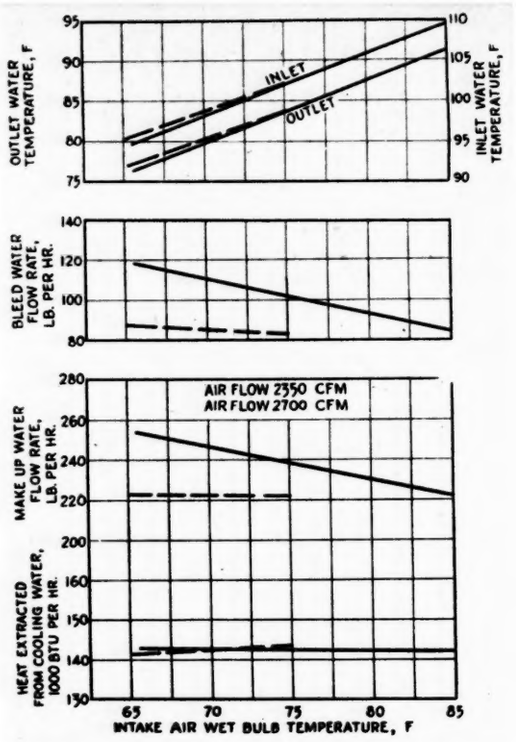

OUTLET WATER TEMPERATURE, F
INLET WATER TEMPERATURE, F
INLET
OUTLET
BLEED WATER FLOW RATE, LB. PER HR.
MAKE UP WATER FLOW RATE, LB. PER HR.
AIR FLOW 2350 CFM
AIR FLOW 2700 CFM
HEAT EXTRACTED FROM COOLING WATER, 1000 BTU PER HR.
INTAKE AIR WET BULB TEMPERATURE, F

Fig. 17. Effect of intake air wet bulb temperature on performance of Servel TB-15 water cooling tower.

In general, air flow through the water cooling tower may be varied by as much as ± 15% from a normal air handling rate of 2,380 cfm without appreciable effect on cooling capacity or make-up water consumption.

Safety and Air Resistance

Winter air conditioning portions of both the 3- and 5-ton conditioners have been tested and found to comply with American Standard Approval Requirements for Central Heating Gas Appliances, and American Gas Association approval has been granted. These units are entitled to display the Laboratories Approval Seal with the following statement on the name plate: "Approved as forced air heating equipment only." There are no American Standard Requirements for gas-fired summer air conditioners. However, since winter air conditioning portions of these approved units include the gas-fired steam boiler, which also supplies steam to the summer air conditioning section, it is quite evident that both performance and construction of this boiler may be considered adequate from the standpoint of safety, durability, and reasonable efficiency. The steam boiler operates at practically atmospheric pressure and is further protected by a steam vent switch which shuts off the gas if all steam generated is not utilized. The sealed refrigerating system was also found to be sturdily constructed and its operation was effectively controlled by automatic devices. Both refrigerant (water) and solvent (lithium bromide solution) are non-toxic and non-inflammable and therefore comply with refrigeration safety codes. Since the refrigeration cycle necessitates the maintenance of a high vacuum and will cease to operate if vacuum is broken there is no hazard from high pressure leakage or explosion. Other than the steam diverter valve and condensate return pump attached to the boiler there are no moving parts.

Resistance of air filters, cooling coil, and heating coil to air flow static pressures was measured in inlet air openings and outlet air ducts of the 3- and 5-ton units during cooling capacity tests with air flow varied from normal to approximately 25± below to 25% above normal. At normal air deliveries of 1,219 and 2,060 cfm for the 3- and 5-ton units, pressure drops of 0.44 and 0.51 in. water column were recorded. Such performance would be considered as normal for equipment of this type.

Williams' Performance

As the Williams refrigeration unit is designed to supply a quantity of chilled water it was necessary to secure the manufacturer's recommendations on chilled water temperature and flow rates as well as to select a chilled water coil to conduct a standard A.S.R.E. air conditioning rating test. Results from this simulated A.S.R.E. test are presented in Table 2.

Data presented in Table 2 and in subsequent sections on the Williams unit show a practical application using chilled water to condition air. The results of these tests are for the chilled water coil and blower system utilized in this investigation which are not a part of the refrigerating unit. As all test results computed from air data agreed to within 3.5% with those computed from chilled water temperatures and flow rates, all results given for this unit may be considered as applying to both air and chilled water data. However, those presented were computed from air data because air measurements yielded additional information on sensible and latent cooling effects achieved with the chilled-water coil employed with this unit, which may be considered a typical selection of these necessary auxiliaries.

Motor requirements consisted of refrigerant pump 701 watts, chilled water pump 470 watts, cooling tower water pump 1,700 watts, cooling tower water fan 491 watts, and air blower 746 watts. If an atmospheric cooling tower were employed, the total electricity consumed would be reduced to 3,617 watts.

As the automatic shut-off switch used on the Williams unit would not permit it to operate continuously with cooling tower water temperatures specified by A.S.R.E., the manufacturer suggested the temperatures listed in Table 2. In effect these were 10F higher than A.S.R.E. standard and for that reason the unit was not operating under as favorable conditions as it would be entitled to. Subsequent tests under other conditions indicated that the Williams would develop more than 7 tons cooling capacity. Under these circumstances it is felt that the manufacturer is fully justified in using a nominal 7-ton rating on this unit, even though test data in Table 2 indicate a rating of 6.78 tons. Reference to Fig. 12 will also indicate that outlet test air was cooled approximately 10F lower than the Servel, indicating that the air quantity handled could have been consider-

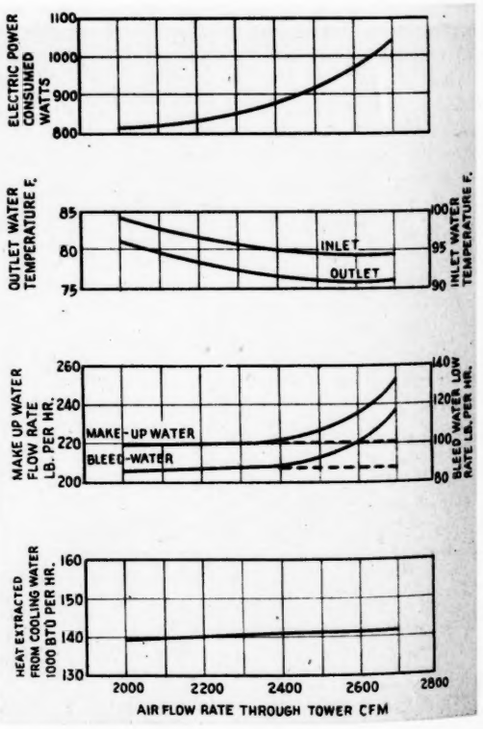

ELECTRIC POWER CONSUMED WATTS
OUTLET WATER TEMPERATURE F.
INLET WATER TEMPERATURE F.
INLET
OUTLET
MAKE UP WATER FLOW RATE LB. PER HR.
MAKE-UP WATER
BLEED-WATER
BLEED WATER FLOW RATE LB. PER HR.
HEAT EXTRACTED FROM COOLING WATER 1000 BTU PER HR.
AIR FLOW RATE THROUGH TOWER CFM

Fig. 18. Effect of variation of air flow through Servel TB-15 water cooling tower on its performance.

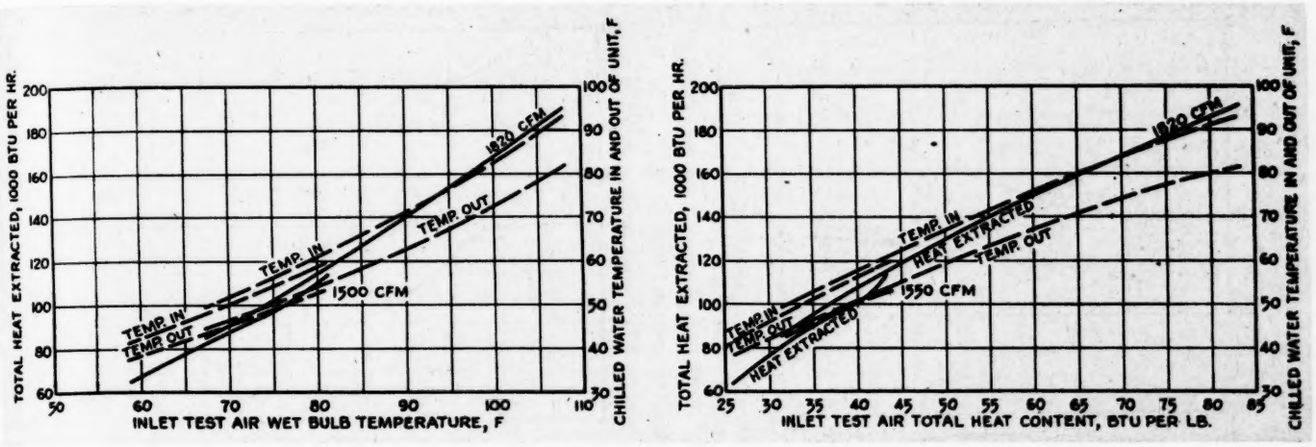


Fig. 19. Effect of increasing wet bulb temperature and total heat content of inlet air on performance of Williams unit.

ably increased. As mentioned previously, this was an arbitrary selection as it was not necessary to discharge the air from the cooling coil at such a low temperature.

Varying Inlet Air Conditions

To determine the operating characteristics of the Williams unit under loads increased far beyond practical conditions, inlet air wet bulb temperature (total heat content) was raised from 58.5F to 108F and air flow rate increased to 1,820 cfm. Results obtained are presented in Fig. 19. As dry bulb temperatures were not maintained at a constant value throughout these tests, data for sensible and latent heat extracted are not included.

When an average air flow rate of 1,820 cfm was employed total heat extracted increased from 85,300 to 189,000 Btu per hr (77.11 to 15.80 tons) as total heat content of inlet air was increased from 31 to 83 Btu per lb (67 to 108F wet bulb). Chilled water temperatures leaving the unit increased from 44.3 to 81.7F while those of water returning to the unit increased from 49.5 to 93.5F. If a chilled water temperature of 42F is required to secure desired dehumidification of air, total heat which could be extracted would be 84,000 Btu per hr (7 tons); if 50F is allowable 100,000 Btu per hr (8 1/3 tons) could be extracted. While chilled water temperatures much above these figures may be considered undesirable for field conditions, these data indicate that the Williams unit continued to maintain a 10-12F differential between water entering and leaving the unit under loads more than double its rated capacity and with no breakpoint far beyond the temperature levels customarily encountered in practice. In other words, its efficiency was maintained with increased loads.

TABLE 2.—TEST DATA OBTAINED ON WILLIAMS 7-TON REFRIGERATION UNIT UNDER STANDARD A.S.R.E. CONDITIONS

Outlet test air dry bulb, F	50.9
Outlet test air wet bulb, F	49.9
Flow rate of air, cfm	1572
Cooling water flow rate, gpm	45.6
Cooling water temp., in, F	89.9
Cooling water temp., out, F	101.1
Chilled water flow rate, gpm	32.7
Chilled water temp., in, F	47.9
Chilled water temp., out, F	42.9
Heat input (steam), Btu per hr.	178,000
Electrical input, watts	4108
Latent heat extracted, Btu per hr.	27,600
Latent heat extracted, tons	2.3
Sensible heat extracted, Btu per hr	53,700
Sensible heat extracted, tons	4.48
Total heat extracted, Btu per hr	81,300
Total heat extracted, tons	6.78

Change in Rate of Air Flow

As a measure of flexibility of the Williams unit to variation in refrigeration load imposed on it, the air flow rate through the cooling coil was varied from 1,180 to 1,965 cfm, 25% below and 25% above normal used for the A.S.R.E. test. Results obtained as presented in Fig. 20 are also dependent on performance of cooling coil and accessory equipment as well as the Williams unit itself. They are given here merely as an indication of expected performance of the Williams unit when similarly installed elsewhere.

By increasing air flow rate 25% above normal, total heat extracted was increased 6.77% from 81,300 to 86,800 Btu per hr and achieving this increase, chilled water outlet temperature increased by only 2.1F, from 42.9 to 45.0F. If higher chilled water temperatures were employed, larger quantities of air could be satisfactorily treated under essentially identical dry and wet bulb temperature conditions, and the cost of refrigeration would be decreased by virtue of improved operating efficiency. In spite of the fact that latent heat extraction was materially reduced at air flow rates greater than 1,630 cfm, outlet air wet bulb temperature obtained with an air flow of 1,965 cfm was only 52.4F, which should be adequate for comfort air conditioning.

Cooling Water Variations

Cooling water was supplied to the Williams unit from a water cooling tower and circulating pump installed as accessory equipment. Results obtained by varying the cooling water temperature or the water flow rate are independent of equipment used and therefore are directly applicable to field conditions.

Data obtained by varying water temperature alone with flow rate maintained constant at an average value of 45.7 gpm are shown in the lower left portion of Fig. 21. Latent and sensible cooling effects were generally decreased as tower water temperature was increased. Irregularities in these trends may be attributed to experimental difficulty in maintaining constant all other variables, such as air and water flow rates or temperature differentials.

As shown on the right side of Fig. 21, increasing tower flow rate from 25.2 to 50.4 gpm with inlet temperature maintained constant at approximately 90F resulted in increasing quantities of total and latent heat being extracted from circulated air. Sensible heat characteristics were relatively unaffected. The fact that an increase of about 5 gpm at the upper end of total heat extracted curves show very little effect indicates that the manufacturer's recommended flow rate of 45 gpm was well chosen.

In localities where cooling water at temperatures below 80F is available, it is possible to operate the Williams unit continuously with less

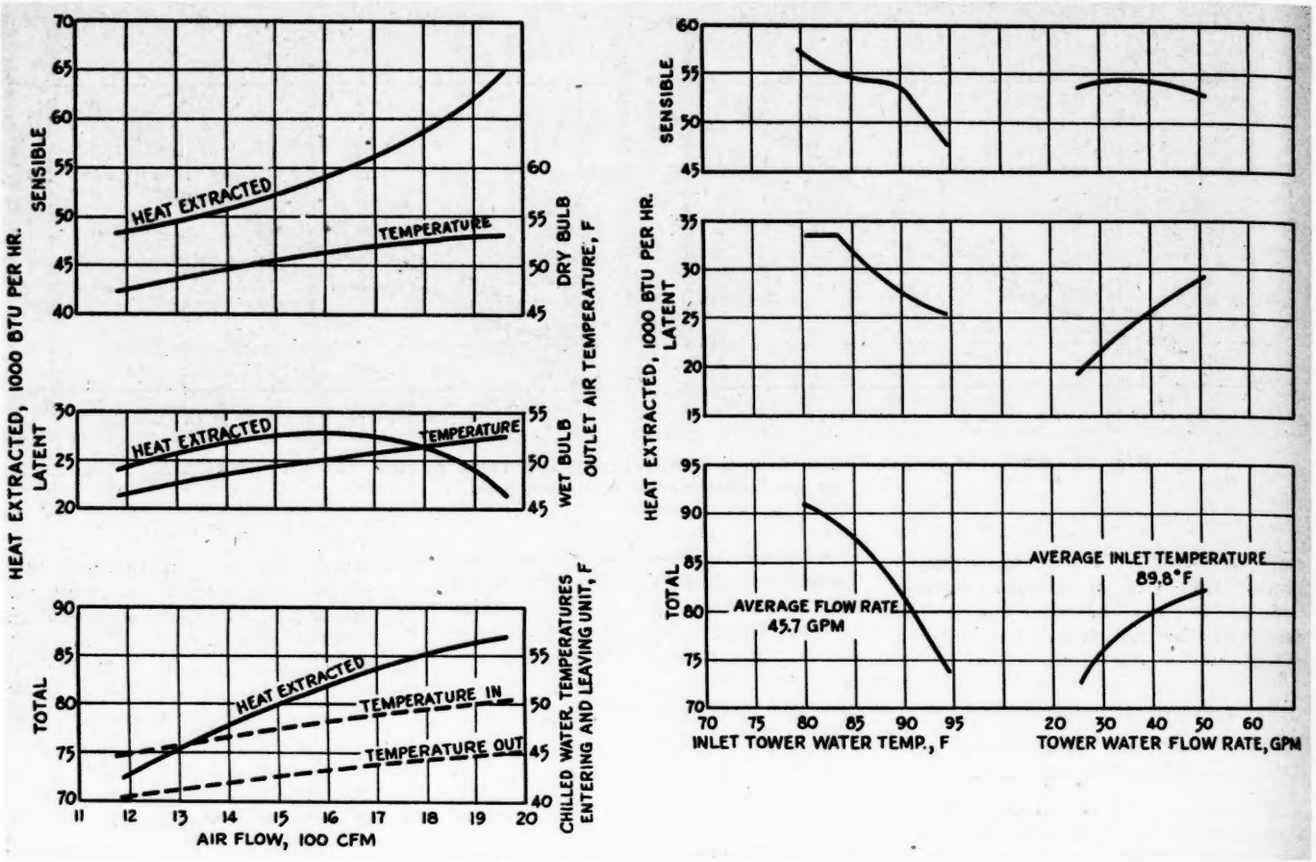


Fig. 20. Effect of air flow rate on performance of Williams 7-ton unit. Fig. 21. Effect of cooling water temperature and flow rate on performance of Williams 7-ton unit.

flow than 45 gpm, as recommended by the manufacturer for 90F water. This possibility was indicated by additional tests in which both temperature and flow rate of cooling water were varied simultaneously. Although results of these tests cannot be directly correlated in the usual manner with those illustrated in the lower portion of Fig. 21, they provide some interesting information. In two tests, tower water inlet temperatures of 70 and 80F were employed in developing cooling capacities of 82,000 and 81,000 Btu per hr, which were quite comparable to 81,300 Btu per hr obtained in the standard rating test. In achieving these capacities, chilled water temperatures and flow rates were maintained at approximately 40-45F and 32.8 gpm as in the standard rating test. Tower water outlet temperatures were adjusted to 107.6 and 100.8F, for inlet temperatures of 70.4 and 80.6F, respectively, by throttling water flow rates to 14.2 and 25.9 gpm. Thus, with a constant cooling capacity of 6.8 tons and a chilled water temperature rise of about 5F, comparisons with the standard rating test may be made only in so far as tower water flow rate and temperature rise are concerned.

From these comparisons it is apparent that reduction of cooling water flow rate by as much as two-thirds does not seriously affect heat transfer coefficients or total cooling capacity so long as cooling water outlet temperatures do not exceed 108F, as indicated by maximum outlet temperature attained in the test conducted with 70.4F inlet water.

Effect of Steam Pressure

A steam temperature of about 243F is recommended by the manufacturer for evaporating refrigerant from solvent in the heating section of the Williams 7-ton unit. Since temperature of steam is a function of pressure, this means that a pressure of approximately 26.5 lb per sq in. absolute (11.8 lb per sq in. gage) would be required for normal operation. To determine effects of variation in steam pressure on performance of the Williams equipment the absolute steam pressure was varied from 22.0 to 29.6 lb per sq in. equivalent to temperatures ranging from 233 to 249.5F.

Total heat extracted with a steam pressure of 22.0 lb per sq in. (absolute) was only 75,450 Btu per hr (6.29 tons). At 24.7 lb per sq in. a total of 77,000 Btu per hr (6.42 tons) was removed, and at 26.5 lb per sq in. (normal condition) 81,300 Btu per hr (6.78 tons). Similar trends were recorded for both latent and sensible heat extracted. It is apparent that the manufacturer has selected the most efficient steam pressure for operation of this unit and best results will be obtained at 26.5 lb per sq in. absolute.

Varying Chilled Water

Temperature and rate of flow of chilled water have a very appreciable and direct effect on cooling capacity of the Williams unit. It is necessary to circulate the chilled water rapidly enough to utilize the full cooling effect developed by the refrigerant, and the temperature of the chilled water must be low enough properly to cool and dehumidify air passed over the cooling coil.

To evaluate refrigerating capacity of the Williams unit under variable loads, primary importance should be attached to chilled water inlet temperatures, outlet temperatures, and total heat extracted from air. In these particular tests, variation in load was secured by increasing the inlet air temperature from 80 to 160F, relative humidities from 20 to 63.5% and air flow rates from 1,207 to 1,823 cfm.

Variations observed in total heat extracted, and chilled water inlet temperatures at various chilled water outlet temperatures are graphically illustrated in the lower part of Fig. 22. It may be noted that as chilled water outlet temperature was increased from 37.77 to 81.65F, total heat extracted was also increased in very nearly direct proportion from 64,400 to 189,600 Btu per hr (5.37 to 15.80 tons).

Since the Williams unit produces chilled water for utilization in cooling and dehumidifying air, any data presented on air conditioning functions are dependent on characteristics of cooling coils employed for reducing temperature of air passing over them as well as those of the blower system used for circulating air over these coils. The particular bank of coils employed in the test rack (a fixed counterflow type heat exchanger adequate for 7 tons of refrigeration) proved to be especially suited for use with this unit, as indicated by the fact that outlet air wet bulb temperatures were within a range of 0.16 to 2.9F of chilled water inlet temperatures. Relative humidities of outlet air were well above 90% in all cases except one in which most of the cooling load was sensible heat.

Manufacturer's recommended operating conditions called for chilled water circulation at a rate of 32.5 gpm. This recommended flow rate was approximated during the first standard test run. To determine effects of chilled water circulation on cooling capacity characteristics, however, three additional tests were conducted in which all other operating conditions were maintained essentially constant, and chilled water flow rate was varied from 19.8 to 33.5 gpm. This latter rate was maximum flow obtainable with the circulating pump employed for this purpose.

Data obtained from these tests are illustrated in the upper portion of Fig. 22, where total, latent and sensible heat extracted are plotted against chilled water flow rate. When chilled water flow rate was reduced from 32.7 to 26.0 gpm, total heat extracted was decreased from 81,300 to 77,800 Btu per hr (6.78 to 6.48 tons). A further decrease in water rate to 19.8 gpm apparently had no effect, since cooling capacity remained constant at 6.48 tons. When chilled water flow rate was increased from 32.7 to 33.5 gpm a slight increase was observed in total heat extracted. At this higher flow rate total heat extracted was 82,100 Btu per hr (6.84 tons).

It is believed that latent and sensible heat trends indicated in Fig. 22 were due to change in chilled water inlet and outlet temperature differential rather than to change in flow rate. To obtain a true comparison, all variables other than chilled water flow rate and sensible heat extracted should decrease as indicated by dotted lines.

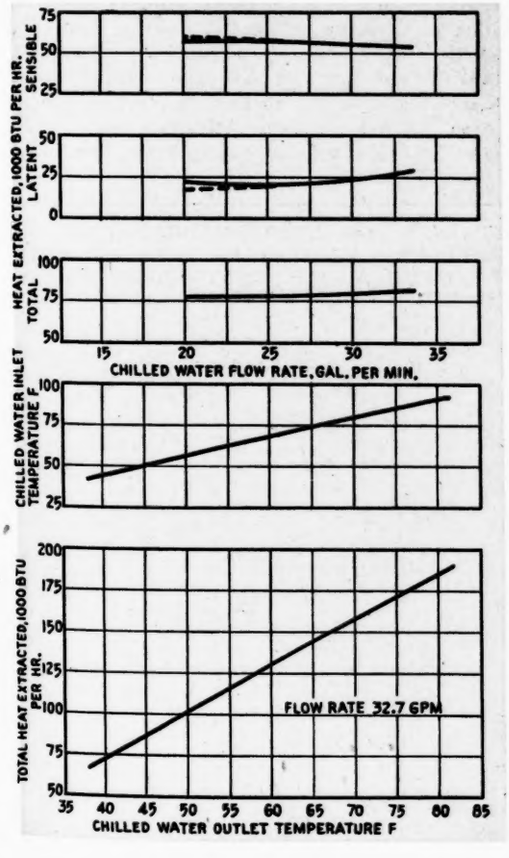

HEAT EXTRACTED, 1000 BTU PER HR. SENSIBLE
LATENT
TOTAL
CHILLED WATER FLOW RATE, GAL. PER MIN.
CHILLED WATER INLET TEMPERATURE F
TOTAL HEAT EXTRACTED, 1000 BTU PER HR.
FLOW RATE 32.7 GPM
CHILLED WATER OUTLET TEMPERATURE F

Fig. 22. Effects of chilled water outlet temperature and flow rate on performance of Williams 7-ton unit.

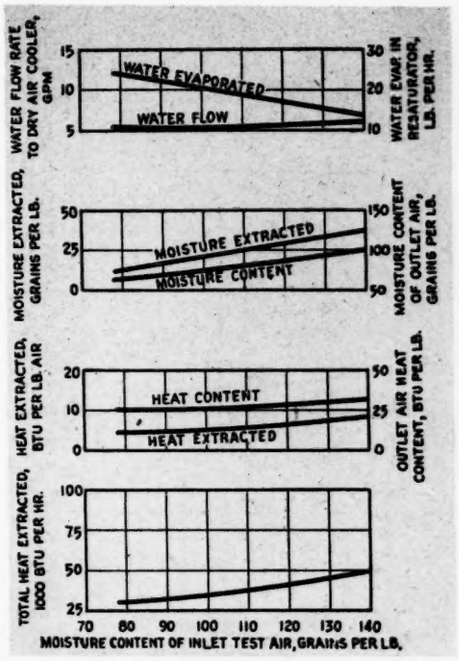

WATER FLOW RATE TO DRY AIR COOLER, GPM
WATER EVAPORATED
WATER FLOW
WATER EVAP IN REGENERATOR, LB. PER HR.
MOISTURE EXTRACTED, GRAINS PER LB.
MOISTURE EXTRACTED
MOISTURE CONTENT
MOISTURE CONTENT OF OUTLET AIR, GRAINS PER LB.
HEAT EXTRACTED, BTU PER LB. AIR
HEAT CONTENT
HEAT EXTRACTED
OUTLET AIR HEAT CONTENT, BTU PER LB.
TOTAL HEAT EXTRACTED, 1000 BTU PER HR.
MOISTURE CONTENT OF INLET TEST AIR, GRAINS PER LB.

Fig. 23. Effect of moisture content of inlet test air on performance of Bryant No. 14R assembly.

Safety

Although the Williams unit utilizes steam from any source provided it is maintained at proper pressure and temperature, it is particularly adaptable to steam generated in gas boilers. Where such gas boilers are used, the A.G.A. recommends that they should comply fully with American Standard Approval Requirements for Central Heating Gas Appliances and the A.S.M.E. Boiler Code. It is important that the required pressure control and relief valves be used on the steam boiler as such devices are not incorporated in the Williams unit.

Refrigerant and solvent are maintained under vacuum of 15-20 in. mercury column when idle. If atmospheric pressure were ever to occur in the Williams unit when not operating, it would indicate that a leak of air inward had developed that would not only cause loss of vacuum, but would also serve to release any excess pressure which might be created if an attempt were made to operate the system. Furthermore, the inner portion of heater section in which highest refrigerant pressures are developed is surrounded by a steam jacket in which an absolute pressure of 27-28 lb is normally maintained. This leaves a differential of only 5 lb per sq in. between the pressures in the internal and external chambers. As the heater section is normally operated under a positive pressure of 1 to 3 lb per sq in. (gage) and subjected to a temperature of approximately 243F, it seems evident that the Williams unit complies with applicable requirements for refrigerant containing parts.

At the conclusion of tests on the Williams unit a sample of solvent and refrigerant was withdrawn from it to determine the effect of the organic chemicals on the metal parts. When tested for acidity the sample was neutral to litmus paper. Likewise, presence of ferric chloride could not be detected with potassium hydroxide using phenolphthalein indicator and potassium ferro-cyanide and potassium thio-cyanate, very sensitive qualitative indicators for presence of iron. These determinations clearly indicated no reaction of hydrochloric acid with the iron of the Williams unit or hydrolysis of either solvent or refrigerant. This agrees with the manufacturer's statement.

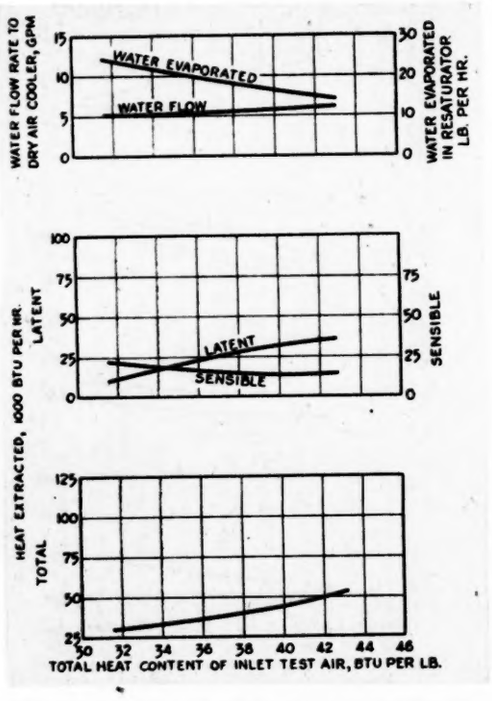

WATER FLOW RATE TO DRY AIR COOLER, GPM
WATER EVAPORATED
WATER FLOW
WATER EVAPORATED IN RESATURATOR LB. PER HR.
LATENT
SENSIBLE
HEAT EXTRACTED, 1000 BTU PER HR.
TOTAL
TOTAL HEAT CONTENT OF INLET TEST AIR, BTU PER LB.

Fig. 24. Effect of total heat content of inlet test air on performance of Bryant No. 14R assembly.

Further information on the refrigerant and solvent may be of interest particularly as it has a bearing on safety requirements. This refrigerant, methylene chloride (dichloromethane), CH_2Cl_2, is in common use as a refrigerant in electric compression refrigeration units. It is a heavy (1.33 sp gr) colorless, mobile liquid, has a pleasant odor, and boils at 103.6F. According to the manufacturer, E. I. duPont de Nemours & Co., Inc., it possesses great thermal stability even at temperatures up to 248F, which is considerably higher than its boiling point. It is rather insoluble in water and small quantities of water do not appreciably affect it below 212F. Methylene chloride, according to Underwriters' Laboratories report, is practically non-inflammable and non-explosive at ordinary temperatures. It is toxic but less so than chloroform or carbon tetrachloride.

The solvent $CO_3O(CH_2CH_2O)_4CH_3$ is a derivative of ethylene glycol. It was first produced in 1937 by Zellhoefer and Marvel. The solubility of methylene chloride in this liquid is very great and is even considered abnormal.

It is believed that with proper precautions in controlling temperatures to which refrigerant and solvent are subjected (248F or below) and thorough drying of the charge of solvent and refrigerant it would be expected to give reasonably long service. No conditions are now apparent in the Williams unit which, if these precautions are followed in detail, should cause any breakdown of refrigerant and solvent liquids, or any reaction of these materials with metals employed in its construction.

Bryant Unit Performance

Although the A.S.R.E. standard rating conditions are not directly applicable to the Bryant assembly (including dehumidifier, dry air cooler, and resaturator) a test was conducted under conditions as near as possible to those specifications. Manufacturer's operating instructions for the Bryant 14R assembly specify a cooling water rate of approximately 8 gpm. This recommended flow rate was not employed in the standard rating test since a flow of only 5.68 gpm produced the required rise of 20F for 75F inlet water. Results of the standard test conducted as nearly as possible according to the A.S.R.E. code are included in Table 3 together with results of a similar test conducted with a cooling water flow rate of 8.1 gpm as recommended by the manufacturer.

It is evident from the data and from a contemplation of the principle on which this system works that A.S.R.E. standard test conditions may unduly handicap dehumidification equipment in which cooling of air depends entirely upon availability and temperature of natural water supply or upon water capacity of a cooling tower as well as on manufacturer's designed capacity for water circulation.

Reactivation Air

Air for reactivation of the moist silica gel is drawn into the Bryant 14R dehumidifier at a rate of approximately 1,000 cfm. For this purpose air may be drawn either from outside the building through a duct or from the room in which the dehumidifier is located. These alternate methods of securing reactivation air were originally considered to have a bearing on the method of conducting a standard rating test because the A.S.R.E. standard method specifies outside air at 95F dry bulb and 75F wet bulb, while equipment room air is supposed to be 80F dry bulb and 67F wet bulb. To observe the effect of variation in wet and dry bulb temperatures of reactivation intake air on overall performance of the Bryant assembly, tests were conducted with this air maintained at dry bulb temperatures ranging from 76.8 to 113.1F and wet bulb temperatures from 66.3 to 88.8F. All other operating conditions were maintained at constant values approximating those of the standard A.S.R.E. test.

Although moisture content of reactivation intake air was varied from 78 to 170 grain per lb, moisture content of conditioned air and total heat extracted remained essentially constant at average values of 68 grains per lb and 29,400 Btu per hr. These results indicate that within the range of comfort air conditions

TABLE 3.—TEST DATA OBTAINED ON COOLING CAPACITY OF BRYANT NO. 14R DEHUMIDIFIER, DRY AIR COOLER, AND RESATURATOR ASSEMBLY UNDER STANDARD A.S.R.E. CONDITIONS AND AT MANUFACTURER'S RECOMMENDED COOLING WATER FLOW RATE

Data Recorded*	A.S.R.E. Standard Method	Manufacturer's Recommended Water Flow Rate
Outlet air, dry bulb, F	66.6	63.3
Outlet air, wet bulb, F	60.1	57.3
Outlet air, moisture content, gr per lb	67	61
Outlet air, heat content, Btu per lb	26.42	24.61
Outlet air, moisture extracted, gr per lb	11	17
Outlet air, heat extracted, Btu per lb	5.28	6.85
Water to dry air cooler, flow rate, gpm	5.68	8.10
Water to dry air cooler, inlet temperature, F.	75.0	74.2
Water to dry air cooler, outlet temperature, F	95.5	89.8
Water evaporated in resaturator, lb per hr.	24.65	22.20
Gas input, Btu per hr	146,000	147,800
Electrical input, Btu per hr	3.434	3.434
Total heat extracted, Btu per hr	30,050	38,800
Latent heat extracted, Btu per hr	9,000	14,230
Sensible heat extracted, Btu per hr	21,050	24,570

*Data not listed conform to standard conditions. Test air flow rate 1,300 cfm, reactivation air flow rate 1,000 cfm.

normally encountered, latent heat and moisture content of reactivation intake air has little or no effect on the reactivation of silica gel. In other words, performance characteristics of the dehumidifier remain essentially constant with variations in reactivation air. This information indicated that temperature and moisture content of reactivation air did not have to be confined within narrow limits during ensuing cooling capacity tests.

Variations of Inlet Air

With other operating conditions maintained at A.S.R.E. standard, a series of cooling capacity tests was conducted with the Bryant assembly in which effects of varying total heat and moisture content of inlet test air were observed.

In Fig. 23 the results of tests with varying moisture content of inlet air on total heat extracted, moisture removal, water flow rate to dry air cooler (to maintain 75F inlet, 95F outlet), and water evaporated in resaturator are illustrated.

To show the effect of varying total heat content of inlet test air on performance, the tests described above were replotted in Fig. 24. In general these curves follow trends similar to those shown in Fig. 23, with total heat content ranging from 31.70 to 43.00 Btu per lb for inlet test air and with total heat extracted, water flow to dry air cooler, and water evaporated in resaturator identical to those shown in Fig. 23. It will be observed that latent heat extracted increased from 9,000 to 36,000 Btu per hr but that sensible heat extracted decreased from 21,050 to 13,700 Btu per hr.

Tests were also conducted with variable dry and wet bulb temperatures of inlet test air but with a fairly constant moisture content averaging 62 grains per lb. Results of these tests are illustrated in Fig. 25, with dry bulb temperature of inlet test air employed as the variable. With dry bulb temperature increased from 80.9 to 106.7F total and sensible heat extracted, heat extracted per pound of air, and water flow rate to dry air cooler increased proportionately, while heat content per pound of air and latent heat extracted (well under 1,000 Btu per hr) remained essentially constant. Total heat extracted increased from 23,700 to 62,300 Btu per hr (1.98 to 5.19 tons); sensible heat extracted was similarly increased from 23,700 to 62,100 Btu per hr, heat extracted per lb of air was raised from 4.14 to 10.91 Btu, and water flow to dry air cooler necessary to maintain standard conditions (75F inlet, 95F outlet) was increased from 4.80 to 7.41 gpm.

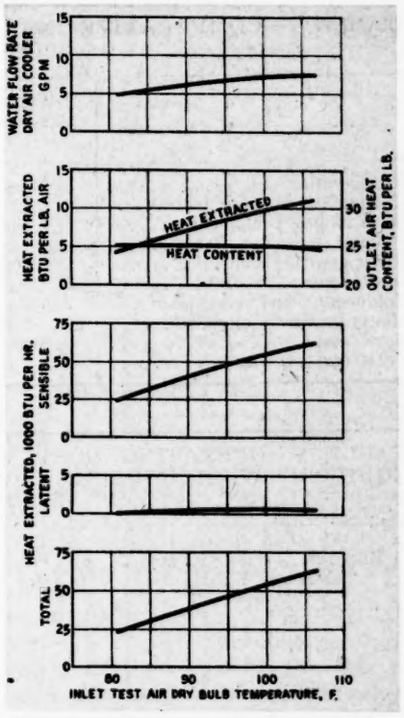
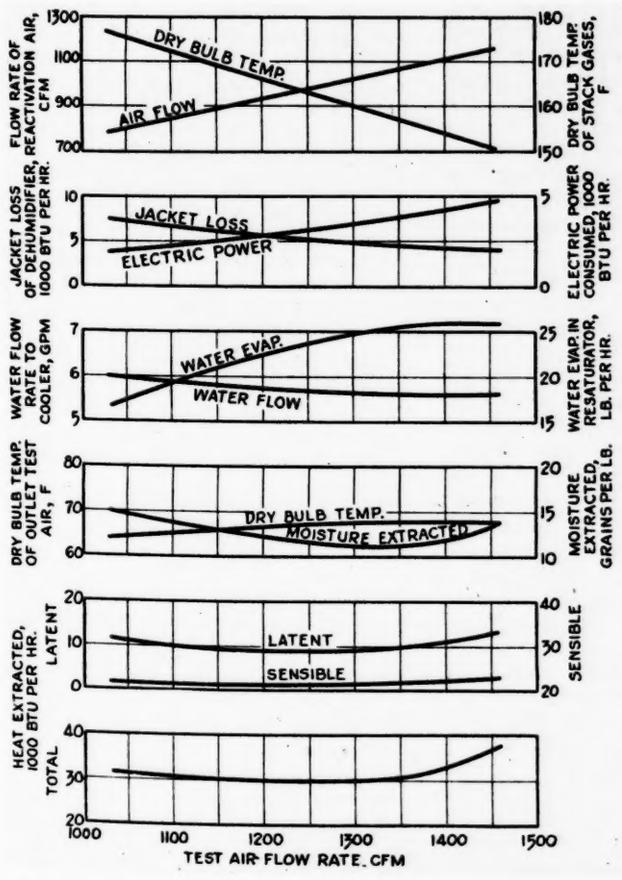

WATER FLOW RATE DRY AIR COOLER GPM
HEAT EXTRACTED BTU PER LB. AIR
HEAT EXTRACTED
HEAT CONTENT
OUTLET AIR HEAT CONTENT, BTU PER LB.
HEAT EXTRACTED, 1000 BTU PER HR. SENSIBLE
HEAT EXTRACTED, 1000 BTU PER HR. LATENT
TOTAL
INLET TEST AIR DRY BULB TEMPERATURE, F.

Fig. 25. Effect of varying dry bulb temperature of inlet test air with constant moisture content on performance, Bryant 14R assembly.


FLOW RATE OF REACTIVATION AIR, CFM
DRY BULB TEMP.
AIR FLOW
DRY BULB TEMP. OF STACK GASES, F
JACKET LOSS OF DEHUMIDIFIER, 1000 BTU PER HR.
JACKET LOSS
ELECTRIC POWER
ELECTRIC POWER CONSUMED, 1000 BTU PER HR.
WATER FLOW RATE TO COOLER, GPM
WATER EVAP.
WATER FLOW
WATER EVAP. IN RESATURATOR, LB. PER HR.
DRY BULB TEMP. OF OUTLET TEST AIR, F
DRY BULB TEMP.
MOISTURE EXTRACTED
MOISTURE EXTRACTED, GRAINS PER LB.
HEAT EXTRACTED, 1000 BTU PER HR. LATENT
LATENT
SENSIBLE
HEAT EXTRACTED, 1000 BTU PER HR. TOTAL
TEST AIR FLOW RATE, CFM

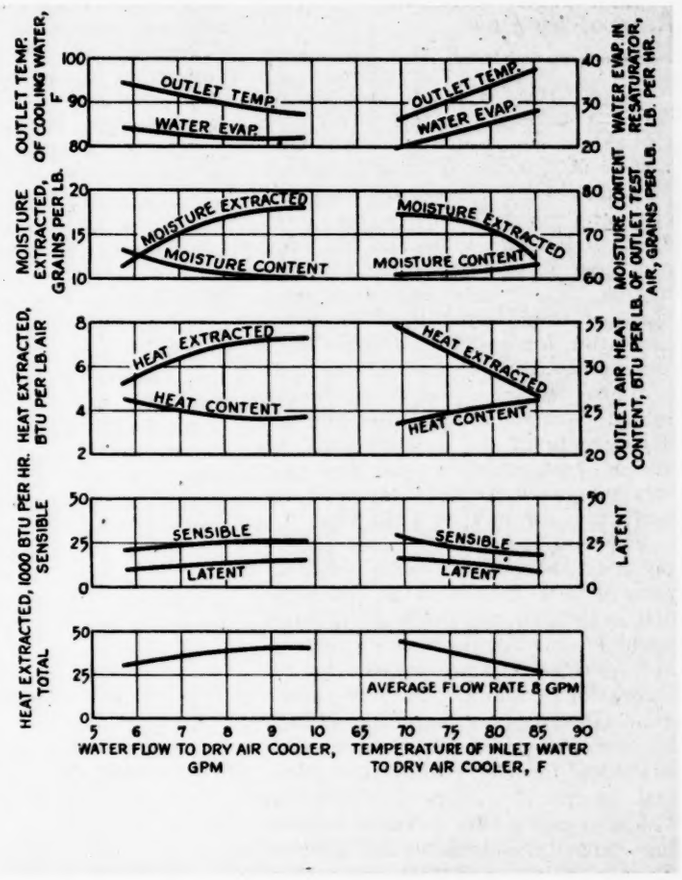

OUTLET TEMP. OF COOLING WATER, F
OUTLET TEMP.
WATER EVAP.
WATER EVAP. IN RESATURATOR, LB. PER HR.
MOISTURE EXTRACTED, GRAINS PER LB.
MOISTURE EXTRACTED
MOISTURE CONTENT
MOISTURE CONTENT OF OUTLET TEST AIR, GRAINS PER LB.
HEAT EXTRACTED, BTU PER LB. AIR
HEAT EXTRACTED
HEAT CONTENT
OUTLET AIR HEAT CONTENT, BTU PER LB.
HEAT EXTRACTED, 1000 BTU PER HR. SENSIBLE
SENSIBLE
LATENT
HEAT EXTRACTED, 1000 BTU PER HR. TOTAL
AVERAGE FLOW RATE 8 GPM
WATER FLOW TO DRY AIR COOLER, GPM
TEMPERATURE OF INLET WATER TO DRY AIR COOLER, F

Fig. 26. (Left) Effect of air flow rate on performance of Bryant No. 14R assembly. Fig. 27. (Right) Effect of cooling water inlet temperature and flow rate of performance of Bryant No. 14R assembly.

TABLE 4.—COMPARATIVE INSTALLED COSTS OF GAS AND ELECTRIC AIR CONDITIONING EQUIPMENT (AS COMPILED BY GAS UTILITY ENGINEERS.)

Item	Williams Air-O-Matic 55 Ton	Comparable Electric Unit	Servel 5 Ton	Comparable Electric Unit	Servel 5 Ton	Comparable Electric Unit
Refrigerating unit	$4,466	$3,261	$1,086	$903	$1,060	$785
Cooling tower or equivalent	1,208	991	—	—	270	148
Air handling equipment	1,044	550	477	477	336	336
Piping	1,075	—	50	—	60	50
Electric wiring control	390	850	22	120	50	75
Insulation	400	50	64	64	20	20
Engineering and misc. labor	—	—	173	173	35	35
House heating equipment	440	518	Furnished	325	Furnished	325
Total cost	9,023	6,220	1,872	2,062	1,831	1,774
Total cost per ton	164	113	374	412	366	355

TABLE 5.—OPERATING COSTS OF ELECTRIC COMPRESSOR UNIT EQUIPPED WITH COOLING TOWER AND CHILLED WATER HEAT EXCHANGER.

Electric Power Rate, Cents per Kw-Hr	Power Costs, Cents per Ton-Hr	Total Costs (Elec., Power and Water), Cents per Ton-Hr
1	1.25	1.28
2	2.51	2.54
3	3.76	3.79
4	5.02	5.05
5	6.27	6.30

These data indicate that the Bryant dry air cooler is effective for sensible cooling since the increased heat removal was primarily the result of its action.

Rate of Air Flow

Performance of the Bryant assembly with variation in rate of air flow through it is shown in Fig. 26. One test was conducted with test air flow decreased from 1,300 to 1,028 cfm, and reactivation air flow rate decreased from 1,000 to 787 cfm, representing a decrease of approximately 25% in air handled on both sides of the dehumidifier. Both reactivation air and test air flow rate were altered simultaneously so that pressure balance would be maintained between reactivation and air drying side of the dehumidifier. In another test circulating air flow rate was increased to 1,458 cfm and reactivation air flow to 1,154 cfm.

The data indicate that this assembly may be operated more economically with a decreased air capacity and accompanying decrease in electrical energy requirements and lesser quantity of water required for resaturation cooling. On the other hand, capacity may be increased 23% by increasing air 12%, water consumption for the resaturator 4.5%, and electrical energy consumption 37.6%. Despite this increase in cooling capacity obtainable at greater air flow rates, manufacturer recommends 1,300 cfm as the most practical and economical operating condition for continuous use under field conditions.

Cooling Water Variations

In addition to tests previously described, additional cooling capacity determinations were made with cooling water temperature varied from 69.2 to 85.2F and flow rate from 5.68 to 9.85 gpm, respectively.

Whether cooling water is circulated in a tower-equipped installation or is supplied directly from a well or city water mains, its temperature probably is dependent on that of the source and on climatic conditions. Consequently, trends indicated at right of Fig. 27 are an index of performance that may be expected under various atmospheric conditions and do not necessarily indicate methods for improving performance. As cooling capacity of the Bryant assembly is directly dependent upon performance of the dry air cooler, however, cooling water temperatures available for a comfort cooling installation are of vital importance. As a flow rate of 8 gpm is recommended by the manufacturer, and since it was pointed out in previous discussion that the Bryant assembly developed a greater cooling capacity at this flow rate than at the flow rate of 5.68 gpm used for the A.S.R.E. test, it was decided that the manufacturer's recommended flow rate would be employed in these comparison tests.

A supplementary test conducted with an inlet water temperature of 96.1F indicated that curves shown in Fig. 27 may be extrapolated with reasonable accuracy for temperatures higher than 85F.

Data obtained during tests conducted with water temperatures constant at 75.0F and with its flow rate varied from 5.68 to 9.85 gpm included total, sensible, and latent heat extracted, as well as heat content of air, moisture content of outlet test air, moisture extracted, outlet temperature of cooling water, and weight of water evaporated in resaturator. These are plotted against cooling water flow rates at left of Fig. 27. These curves show that as water flow rate is increased, total, sensible and latent heat extracted, heat extracted per lb of air and moisture extracted are increased while outlet temperature of cooling water and rate of water evaporated in resaturator are reduced.

The data show that it is advantageous to use a cooling water flow rate of 8 gpm as recommended by manufacturer and as low a temperature of cooling water as possible.

Resaturator Shut Off

In many instances the dehumidifier and dry air cooler will be used

TABLE 6.—OPERATING COSTS OF ELECTRIC COMPRESSOR UNIT WITHOUT COOLING TOWER.

Electric Power Rate, Cents per Kw-Hr	Power Cost, Cents per Ton-Hr	Total Cost, Cents per Ton-Hr
1	1.18	2.40
2	2.36	3.58
3	3.54	4.76
4	4.72	5.94
5	5.90	7.12

to provide dehydration and the resaturator will be omitted.

In the A.S.R.E. rating test, with cooling water flow rate of 5.68 gpm, outlet air temperatures were 86.6F dry bulb and 60.1F wet bulb, representing moisture and heat content of 35 grains and 26.42 Btu per lb, respectively. In this instance moisture extracted to 5.28 Btu per lb. Total heat extracted is essentially constant at 30,050 Btu per hr (2.50 tons) whether the resaturator is employed to partially rehumidify the air or not, but latent heat extracted amounted to 35,800 Btu per hr, and sensible heat was added in the amount of 5,750 Btu per hr.

Tests showed that although dry bulb temperature of conditioned air will be substantially higher than that of inlet air in this type of system (without partial rehumidification), latent heat and moisture contents are decreased appreciably. In this kind of service, the silica gel equipment has dehumidification characteristics not readily available with refrigeration systems.

Safety; Power Consumption

The dehumidifier was found to meet applicable approval requirements relating to the construction of gas warm air furnaces. Of the 14 performance requirements for gas furnaces which might be considered applicable to the dehumidifier, all except two were fully met. One of these indicated the desirability of using larger capacity gas controls for gases having a heating value of 800 Btu per cu ft or less. The other indicated advisability of either using a draft diverting device on the reactivation discharge stack or a safety mechanism which would close off the gas supply before incomplete combustion occurred in case the chimney flue were partially blocked.

Most of the electrical energy consumed in operating the Bryant assembly was utilized for driving the 1 hp motor located in the dehumidifier. The only other motor was a 50 watt synchronous conductor type employed for rotating the resaturator screen. Power consumed by the 1 hp motor was dependent on amount of air delivered by circulating fans. Under normal operating conditions the dehumidifier motor and control equipment consumed 1,005 watts, resaturator motor 18 watts, and water valve on dry air cooler 17 watts, totaling 1,040 watts.

TABLE 7.—TOTAL OPERATING COSTS OF SERVEL UNIT WITH COOLING TOWER, CENTS PER TON-HR WITH WATER AT $1 PER 1000 CU FT

Electric Rate, Cents per Kw-Hr	Gas Rate, Cents per Therm				
	2	6	10	14	18
	Operating Costs, Cents per Ton-Hour				
1	0.70	1.50	2.30	3.10	3.90
2	0.95	1.75	2.55	3.35	4.15
3	1.20	2.00	2.80	3.60	4.40
4	1.45	2.25	3.05	3.85	4.65
5	1.70	2.50	3.30	4.10	4.90

INVESTMENT AND OPERATING COSTS

COST of operating, including original investment expense for gas summer air conditioners, depends upon a number of factors and varies widely in different parts of the country. Controlling factors include size and type of unit, building construction, prevailing labor and accessory materials cost, energy and water costs, and local permit and inspection fees. Many of these factors will undoubtedly change under post-war conditions. However, an attempt is made in this section to present some data that may be of interest at this time.

Cost data on 58 residential and small commercial installations of three different makes of gas-fired summer air conditioning units obtained from various gas companies during the last year indicate that for equipment ranging from 1.6 to 20 tons refrigerating capacity, installation costs varied from $300 to $670 per ton, and averaged $460 per ton. These figures do not take into consideration the fact that certain units are combined summer cooling and winter heating units and as such should be credited with the cost of an excellent heating installation. Cost of a heating system, of course, depends on many factors such as design and construction of house, climate of the locality, and type of fuel selected.

Comparative installation costs for gas and electric air conditioning units were also obtained from several gas companies, based on prevailing price levels in their localities, and are shown in Table 4. Data in this table are intended only for individual comparisons with corresponding electrical equipment; they are not intended for comparisons between gas-fired units, since certain items have been omitted approximately equal for both gas and electrical installations. On the average, the three installations of gas equipment show cost about $5 more per ton than all-electric equipment. In one instance, however, installation costs of gas equipment were under those of electric. More favorable installation cost comparisons can be expected when the gas equipment market is expanded so that mass production will be warranted.

Operating costs employed herein are a composite of data from numerous operating field installations, cooling capacity tests conducted at the A.G.A. Testing Laboratories, and

TABLE 8—TOTAL OPERATING COST OF WILLIAMS UNIT WITH COOLING TOWER, CENTS PER TON-HR, WITH WATER AT $1 PER 1000 CU FT

Electric Rate, Cents per Kw-Hr	Gas Rate, Cents per Therm				
	2	6	10	14	18
1	1.07	2.41	3.75	5.08	6.42
2	1.42	2.75	4.09	5.43	6.76
3	1.76	3.10	4.44	5.77	7.11
4	2.11	3.44	4.78	6.12	7.45
5	2.45	3.79	5.13	6.46	7.80

TABLE 9—TOTAL OPERATING COST OF WILLIAMS UNIT WITHOUT COOLING TOWER, CENTS PER TON-HR, WITH WATER AT $1 PER 1000 CU FT

Electric Rate, Cents per Kw-Hr	Gas Rate, Cents per Therm				
	2	6	10	14	18
1	4.04	5.38	6.71	8.05	9.39
2	4.23	5.57	6.91	8.24	9.58
3	4.43	5.76	7.10	8.43	9.77
4	4.62	5.96	7.29	8.63	9.96
5	4.81	6.15	7.49	8.82	10.16

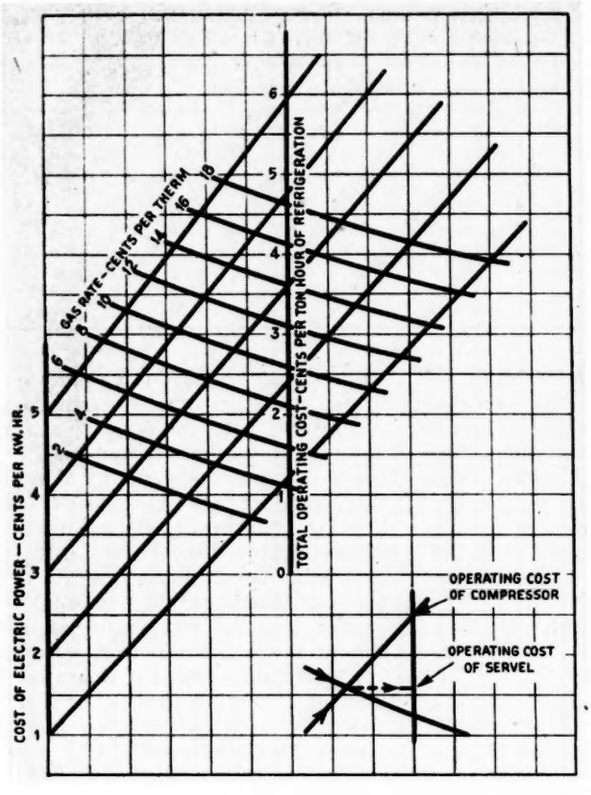

COST OF ELECTRIC POWER – CENTS PER KW. HR.
GAS RATE – CENTS PER THERM
TOTAL OPERATING COST – CENTS PER TON HOUR OF REFRIGERATION
OPERATING COST OF COMPRESSOR
OPERATING COST OF SERVEL

Fig. 28. Comparative operating costs of Servel unit and electric compressor unit with cooling tower.

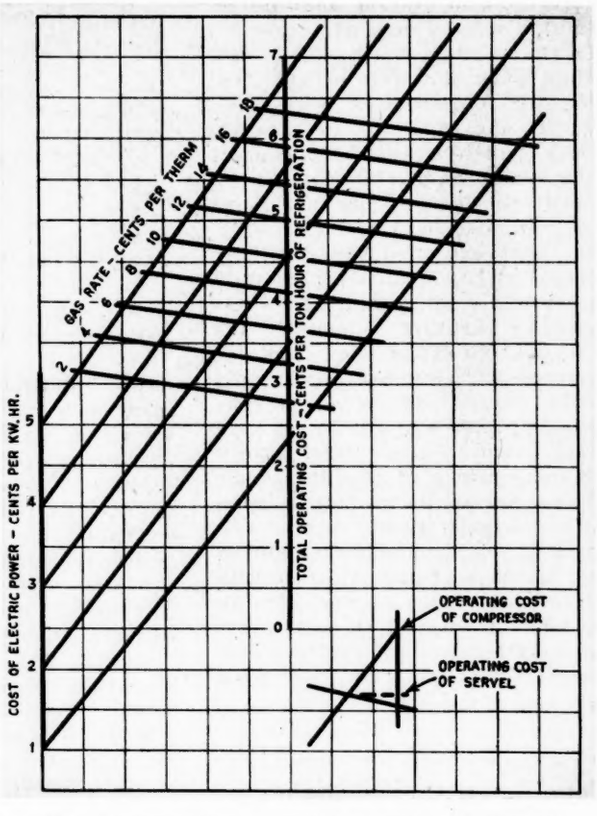

COST OF ELECTRIC POWER – CENTS PER KW. HR.
GAS RATE – CENTS PER THERM
TOTAL OPERATING COST – CENTS PER TON HOUR OF REFRIGERATION
OPERATING COST OF COMPRESSOR
OPERATING COST OF SERVEL

Fig. 29. Comparative operating costs of Servel unit and electric compressor unit without cooling tower.

data supplied by air conditioning equipment manufacturers' specifications. No attempt has been made to include comparisons on depreciation and maintenance costs for different types of equipment as this would unduly complicate studies of operating costs. However, gas utility engineers feel that gas equipment may have distinct advantages along these lines. Various gas equipments for air conditioning differ in principle of operation. For that reason each type of equipment studied is treated individually in comparative studies with electric compression systems.

Operating costs of gas absorption equipment as compared with those of electric compression equipment are influenced primarily by electric power, gas, and water costs. Since gas absorption equipment requires auxiliary electric power for its operation, power cost must be calculated at the same rate which determines the operating cost of comparative electric compression equipment. Energy plus water cost to produce a unit of refrigerating effect are the basic operating costs for the two types of equipment. These are expressed in cents per ton-hour for operating conditions fixed by the A.S.R.E. as standard conditions, namely, inside air 80F dry bulb, 67F wet bulb; outside air 95F dry bulb, 75F wet bulb; condensing water, 75F inlet, 95F outlet.

Electric System Costs

Operating costs of an electric compressor unit equipped with a water cooling tower and chilled water heat exchanger are calculated from the following data:

Compressor power—1.02 kw per ton;
(Seasonal average allowing for cycling losses for moderate size installation).

Auxiliary power:

Condensing-water pump	0.037 hp per ton
Cooling tower fan motor	0.050 hp per ton
Chilled-water pump	0.080 hp per ton
Fan motor	0.100 hp per ton
Total	0.267 hp per ton

Total power—1.254 kw per ton (assuming 85% efficiency for auxiliary equipment).

Water cost, assuming a 2% loss in cooling tower, a 20 degree rise, and water at $1.00 per M cu ft = 0.03 cents per ton-hr.

Operating costs of this electric compressor unit at various power rates are given in Table 5.

The computations for Table 5 were made on the assumption that a cooling tower was provided for condensing-water in the operation of the compressor unit. If a cooling tower is not provided, cooling water must be disposed of, and water consumption becomes equal to the volume of cooling water required rather than that required for make-up with a cooling tower. However, a condensing water pump and cooling-tower fan would not be necessary. Therefore, eliminating the cooling tower reduced auxiliary power costs but increases water costs. The following computations are made on the assumption that a cooling tower is not employed.

Compressor power	1.02 kw per ton
Auxiliary power (.18 hp)	.16 kw per ton
Total Power	1.18 kw per ton

Water costs, at $1.00 per M cu ft, = 1.22 cents per ton-hour.

Operating costs of the electric compressor unit without a cooling tower at various power rates are given in Table 6.

Servel Costs

Operating costs of a Servel absorption unit and an electric compressor unit with both employing a cooling water tower will first be compared. Previous data used for the electric compressor unit remains unchanged except the chilled water pump motor may be eliminated. The total electric energy per ton required equals 1.184 kw-hr, while water costs remain at 0.03 cents per ton-hr.

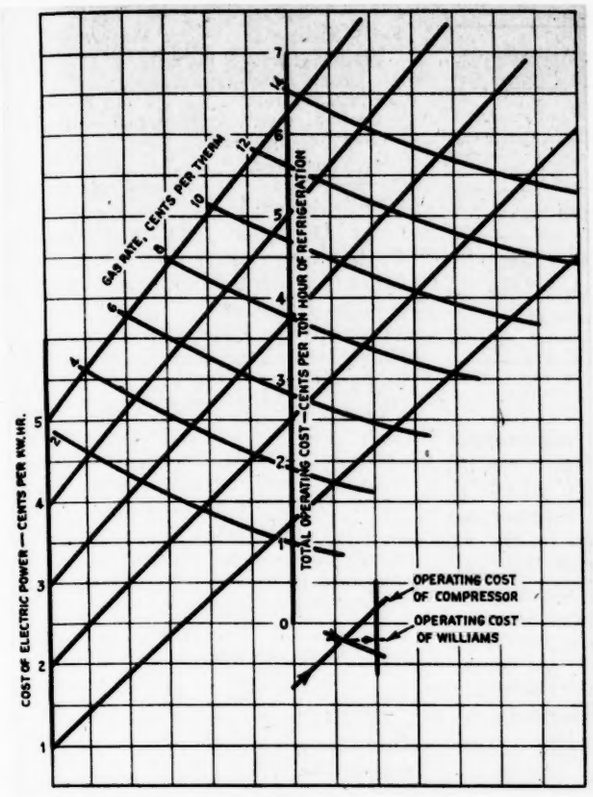

COST OF ELECTRIC POWER — CENTS PER KW.HR.
GAS RATE, CENTS PER THERM
TOTAL OPERATING COST — CENTS PER TON HOUR OF REFRIGERATION
OPERATING COST OF COMPRESSOR
OPERATING COST OF WILLIAMS

Fig. 30. Comparative operating costs of Williams unit and electric compressor unit with cooling tower.

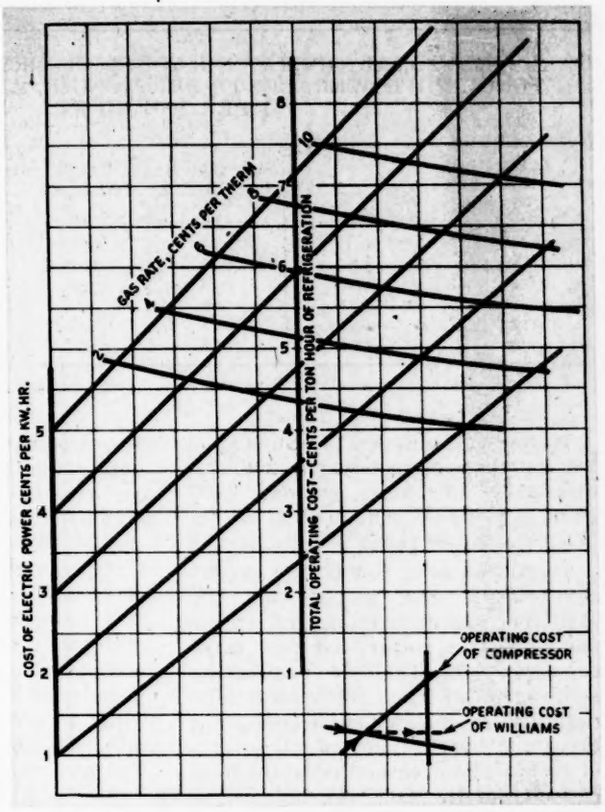

COST OF ELECTRIC POWER CENTS PER KW.HR.
GAS RATE, CENTS PER THERM
TOTAL OPERATING COST — CENTS PER TON HOUR OF REFRIGERATION
OPERATING COST OF COMPRESSOR
OPERATING COST OF WILLIAMS

Fig. 31. Comparative operating costs of Williams unit and electric compressor unit without cooling tower.

The following data were used to calculate operating costs of the Servel Unit:

Fuel Consumption...20,000 Btu per ton-hr.
Auxiliary power:
Fan .....0.100 hp per ton
Cooling tower fan ....0.100 hp per ton
Condensing water pump 0.066 hp per ton
Total .....0.266 hp per ton
= 0.233 kw per ton
Total power, including condensate pump and controls.....= 0.25 kw per ton

Water consumption, assuming a 2% loss in make-up water, a 20F rise, and a 17% flue loss, equals:

0.45 cu ft per ton-hr, which at $1.00 per
M cu ft = 0.045, or approximately 0.05 cents per ton-hr.

Total operating costs of the Servel unit employing a cooling tower are given in Table 7 for various electric power and fuel rates.

Comparative operating costs of the Servel and an electric compressor unit, both employing cooling towers, are presented in Fig. 28. To determine operating cost of the absorption unit at a particular gas rate and electric rate the following method may be employed: Select the curve representing a given gas rate and follow it downward until it intersects a line representing the given electric power rate. From this point of intersection proceed across horizontally to the ordinate representing operating cost in cents per ton of refrigeration and read the operating cost. For example, with a gas rate of 6c per therm and an electric rate of 2c per kw-hr follow the line representing 6c per therm gas rate downward to its intersection of the line representing 2c per kw-hr electric rate. From this point of intersection proceed across horizontally to the operating cost of the electric compressor unit at this given rate of 2c per kw-hr; select the line representing this rate and proceed upward to its intersection of the operating cost ordinate. The operating cost of this unit at 2c per kw-hr is thus given as 2.40c per ton-hr. Comparing operating costs of these two units at a gas rate of 6c per therm and an electric rate of 2c per kw-hr, shows an operating cost of 1.7c per ton-hr for the absorption unit and 2.40c per ton-hr for the compressor unit.

It will be noted that where curves representing gas and electric rates intersect left of the total operating cost ordinate, operating costs of the absorption unit will be less than those of the compressor unit, the difference between the two becoming greater, the greater the distance between the point of intersection and the operating cost ordinate. Likewise, points of intersection at right of the operating cost ordinate indicate that operating costs of the electric compressor unit will be less. It will also be noted that if the point of intersection falls directly on the ordinate line, operating costs of both types of units will be equal. For example, if the electric rate is 2c per kw-hr the operating cost of the compressor unit would equal the operating cost of the absorption unit when the gas rate is 9.20c per therm (curve for this rate not drawn), the operating cost of both being 2.40c per ton.

A comparison of operating costs of the Servel and electric compressor units was also made when neither was provided with a cooling tower. This comparison is represented graphically in Fig. 29. A comparison of Figs. 28 and 29 indicates the advantage of employing a cooling tower in increasing the maximum permissible gas rates which can compete with various electric rates in respect to operating costs.

Williams Costs

Operating costs of a Williams absorption unit employing a cooling tower were calculated from the following data:

Fuel Consumption..33,400 Btu per ton-hr.
Auxiliary Power:
Solution pump .....0.040 hp per ton
Condensing water pump..0.083 hp per ton
Cooling tower fan .....0.090 hp per ton
Chilled water pump .....0.080 hp per ton
Fan motor .....0.100 hp per ton
Total .....0.393 hp per ton
= 0.345 kw per ton

TABLE 10.—TOTAL OPERATING COSTS OF BRYANT SYSTEM WITHOUT COOLING TOWER, CENTS PER TON-HR, AND WITH WATER AT $1 PER 1000 CU FT.

Electric Rate, Cents per Kw-Hr	Gas Rate, Cents per Therm				
	2	6	10	14	18
1	3.38	5.18	6.98	8.78	10.58
2	3.78	5.58	7.38	9.18	10.98
3	4.18	5.98	7.78	9.58	11.38
4	4.58	6.38	8.18	9.98	11.78
5	4.98	6.78	8.58	10.38	12.18

Water consumption (assuming a 2% loss in cooling tower, a 20F rise and 17% flue loss) equals: 0.637 cu ft per ton-hr, which at $1.00 per M cu ft = 0.06 cents per ton-hr.

Total operating costs (fuel, auxiliary power, and water) for the Williams gas absorption unit at various electric power and fuel rates are given in Table 8. Operating costs increase when refrigerant temperature is lowered with resultant greater dehumidifying effect. In other words, the greater the latent heat capacity of the unit, the higher the operating costs will be. This, of course, is also true of electrically-operated compressors.

Data included in Tables 5 and 8 are plotted together in Fig. 30 to present a contrast between operating costs of the Williams gas absorption unit and an electric compressor unit. A water rate of $1.00 per 1000 cu ft was used in calculating data plotted in Fig. 30. This was considered to be a good average rate for all parts of the country. Since an absorption unit requires more condensing water than a compressor unit of equal size, its comparative operating costs become less favorable as water rates increase.

Operating costs of Williams absorption unit without cooling tower are based on the following:

Fuel consumption—
33,400 Btu per ton-hr.

Auxiliary power—
0.193 kw per ton.

Water costs at $1.00 per M cu ft = 3.18 cents per ton-hr.

Total operating costs of the Williams absorption unit without a cooling tower and at various electric power and gas rates are given in Table 9.

Operating costs of the Williams absorption and electric compressor units, tabulated in Tables 6 and 9, are compared in Fig. 31 which is employed in the same manner as Fig. 30. It will be noticed in Fig. 31 that the maximum permissible gas rate at which equal operating costs may be maintained between the two systems, without cooling tower and with an electric rate of 2c per kw-hr is only 0.04c per therm, contrasted with a minimum permissible gas rate of 5.36c per therm when a cooling tower is employed.

Bryant Costs

In the case of dehumidifiers such as the Bryant unit, it is questionable whether output should be expressed in tons of refrigeration. Furthermore, this type of air conditioner is not a refrigeration unit and therefore a comparison between it and the electric compressor can not properly be made. In fact, the dehydrator affords service of an entirely different nature than that performed by compression or absorption units. However, the following data were developed to give a general idea of the performance of a Bryant 14R unit assuming an output of 3.25 tons. The dry air cooler used with the Bryant unit is provided primarily to reduce the temperature of the dried air to the temperature of utilization, which in most cases means the lowest possible point. Since tap water is generally available at temperatures below those obtainable from cooling towers, it is general practice to circulate tap water through dry air coolers. For this reason, cost figures for the Bryant system are given only considering the direct use of city water.

Fuel consumption..45,000 Btu per ton-hr.

Auxiliary Power (blowers, drum drive resaturator and controls) .....0.320 kw per ton

Cooling water consumption without a cooling tower equals:

$$\frac{2.59 \text{ gpm} \times 60}{7.5} = 20.7 \text{ cu ft per ton-hr}$$

At $1.00 per 1000 cu ft, cooling water costs would be 0.04 and 2.07 cents per ton-hr with and without cooling tower, respectively, and assuming (from Laboratories test data) that resaturator water consumption amounts to 22.2 lb per hr, this section required 0.109 cu ft per ton-hr at a cost of 0.01 cents per ton-hr.

Operating costs of the Bryant unit without a cooling tower are tabulated in Table 10. As previously stated, a true comparison of this system with compressor absorption type systems is not possible as they provide different types of service. For this reason no comparison costs have been computed.

Summary

At a cost rate of 2.0c per kw-hr for electricity, gas equipment will compete at 5.36 and 9.20c per therm for the two absorption systems studied when cooling towers are used and with a water rate of $1.00 per 1000 cu ft. Water costs have little effect on the limiting gas rates for any given electric rate when a cooling tower is employed. However, without a cooling tower, gas air conditioning equipment is placed in a much less favorable competitive position due to its greater water requirements. This, however, is not a great disadvantage since the total operating costs of electric compressors increase 66% and 52%, respectively, at electric rates of 1c and 2c per kw-hr when a cooling tower is not used. For all types of systems, operating savings as well as problems of water supply and disposal very generally tend to favor use of a cooling tower.

Thus far equipment and installation costs for gas-fired air conditioning equipment have appeared to be somewhat higher than those for electric compressor systems. However, it is safe to assume that further development of equipment and improvement of installation technique will tend to bring costs of gas equipment in line with the market.

Refrigeration Facilitates Metal Working

F. O. JORDAN

Mechanical Engineer, Albert Kahn Associated Architects and Engineers, Inc., Detroit, Mich.

EVERYONE is familiar with refrigeration's old time role of preserving perishable foods, but the fact and extent of refrigeration's applications to metal working is not common knowledge. Nevertheless, refrigeration is speeding up production and improving products turned out by metal working shops. Industrial refrigeration, after the war, must continue to expand to wider uses.

The principal applications of refrigeration to metal working fall into the following general classes:

Chilling the cooling solutions required to prevent overheating in high speed machining.

Shrinking parts for making expanded pressure fits.

Improving the qualities of metals.

Making possible welding of certain metals.

Cooling the cutting tool and adjacent stock being machined has long been the established practice in machining many metals, especially at high cutting speeds. This is to permit higher cutting rates and to prolong the life of the cutting edge. Various liquids such as cutting oils, and solutions of soap and water are used for this purpose. Solutions are pumped through suitable piping system to be forced against tool and work and afterward returned again for pumping through the system.

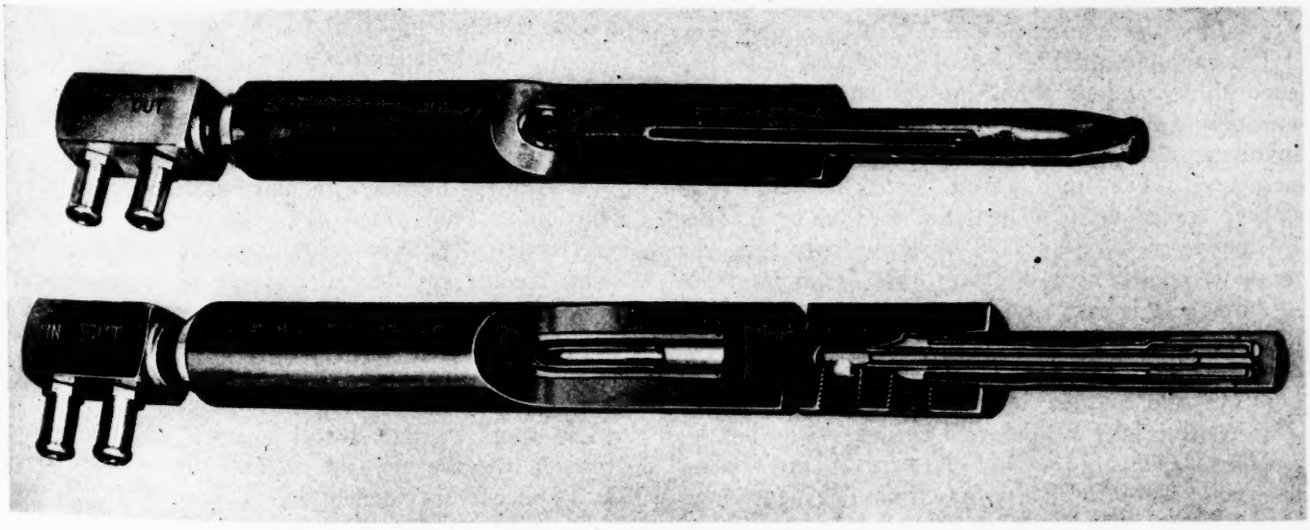
With the increase in harder and tougher alloy metals, and the pressure for stepped up production, the demand for cooling grew. Thus it was a natural step for refrigeration to be brought into the picture.

No particularly new modification in refrigeration machinery was required for this service, as the temperature of the cooling fluid is merely reduced in a conventional liquid cooler. After cooling, it is pumped through the same piping system used before refrigeration was installed. Automatic control is required for maintaining constant fluid temperature to avoid contraction and expansion of the parts being machined and resultant variation in tolerances.

Shrink fits made by heating parts to expand them and then slipping them over some other part and letting them contract by cooling has been done for a long time. The village blacksmith fitted metal rims on wagon wheels by heating the rim in that way. Similarly, tight fits are now made without undesirable effect on the characteristics of the metal by contracting the inside part through refrigeration until it will slip into the other part, and then allowing it to expand by warming to ordinary surrounding temperatures.* This is so simple, the only thing unusual about it is that it was not done a long time ago.

Until recently, airplane engine cylinders with their fins were machined from solid ingots. Obviously, this practice wasted metal and involved considerable machining costs, for much metal had to be machined away between the fins. Now the plain cylinder barrel is cooled to about —40F in a low temperature refrigerated cabinet until the fins slip on readily. When the barrel returns to normal temperature, it expands into the rings so tightly, they

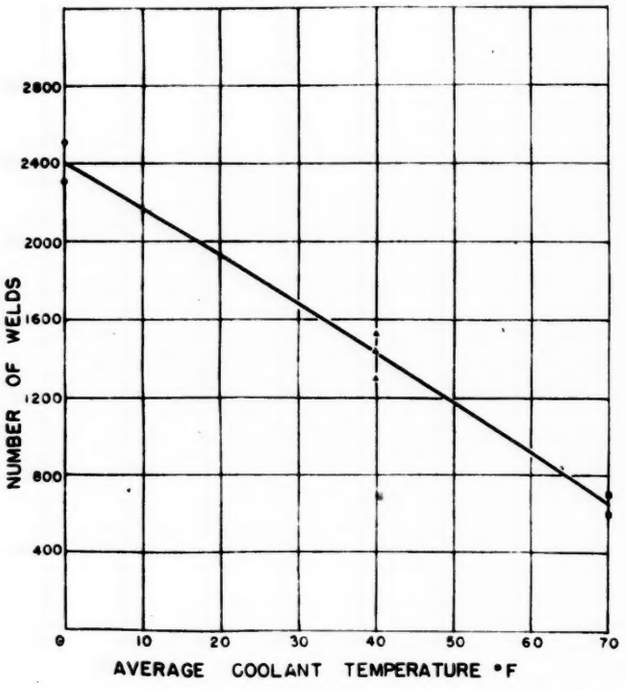
*See "Refrigeration Speeds Shrink Fit Assembly," by F. W. Whitcomb and A. C. Houser, Heating and Ventilating, January, 1944, page 45.



Cut away views of refrigerated electrodes.

might well be all one piece in so far as their effectiveness in discarding the heat of combustion is concerned.

For many years "heat treat" has been an extremely important factor in controlling the hardness and strength of steels. The "village smithy" used it long ago. Until recently, the heat treat has been limited to heating and quenching to various degrees,



Effect of coolant temperature on number of welds made with one type of cooled electrode before cleaning becomes necessary.

at various rates, and by different methods. The blacksmith was adept at this with his forge for heating and his tub of water for quenching. But now, competition would force him to invest in refrigeration of the very low temperature variety.

Modern heat treat includes the use of temperatures as low as — 130F. Hardness, strength, ductility and stability are some of the characteristics improved by low temperature heat treat.

For example, the life of 18-4-1 tool steel is lengthened three or four times due to improvements in hardness and ductility resulting from a treatment involving alternate exposure to high and low temperatures. The steel is first heated to 2300F and is then cooled or quenched in the ordinary manner by means of oil, salt, lead or air. After this, the metal is cooled at a constant rate for 6 hours to — 110F. It is very important that the cooling be done at a constant rate. After the steel has been cooled, it is allowed to warm normally to room temperature and is then tempered at 1000 to 1100F. This routine may be followed through twice.

Alloy steels of 12 to 14% are treated by cooling to — 80 or — 90F, then drawn to 325 degrees to prevent cracking. Nickel carburizing steels treated at — 120F show a gain of 12 to 15 points on the Rockwood "C" scale. Tungsten steels are carburized and given low temperature treatment to improve surface hardness and core strength. Growing of metals is eliminated by changing the austenitic structure to martensitic structure through low temperature treatment. In fact, it may be said that low temperature treatment is a requisite for obtaining a completely drawn micro-structure in tool steels.

Aging of Metal Speeded

Tool steels are not the only metals vastly improved by heat treat at low temperatures. In the past, several years were required for aging steels for precision instruments to permit the metal used to become stabilized. This process that took so long for nature to complete is now taken care of in a couple of days, thanks to refrigeration. The procedure now employed is:

1—Cool metal to — 60F.
2—Warm gradually to 200F and soak for 10 hours in an oil bath.
3—Cool to — 60F for another 10 hours.
4—Warm to room temperature.

The metal is then ready for machining, to be made into precision instruments and tools requiring close and unchanging tolerances.

The low temperature cooling of steels is done either by placing them in refrigerated cabinets, or by immersing them in baths of non-freezing liquids. When cabinets are used, the parts may either be placed in them dry, or immersed in baths of non-freezing liquid. Immersing the parts, it is said, tends to eliminate internal stresses.

Other uses of refrigeration in the machine shop include:

1—Storage of airplane "skins" at sub-zero temperatures.
2—Storage of aluminum rivets and parts at —15F to prevent hardening and also for shrinking.
3—Maintaining quenching tanks at 70F for treatment of shell cases.
4—Maintaining photographic developing baths at constant temperature.

Welding Electrodes Cooled

One of the most spectacular applications of refrigeration to shop practices is the cooling of welding electrodes until they actually are coated with frost. This sounds paradoxical because we know dependable welding is impossible unless metal temperatures are above the level of fusing. However, the refrigerating effect is not applied directly to the surface where the actual welding takes place.

In spot welding metal sheets, the sheets are pressed together by the welding tips or electrodes and the current turned on. The resistance to the flow of electrical current through the surfaces of the two sheets where they are held against each other

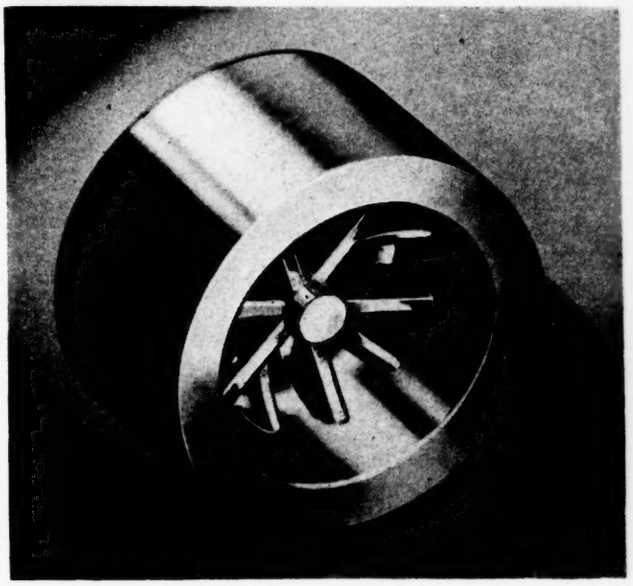
generates the heat at the spot of contact between the two sheets necessary for fusion temperatures and welding. Refrigeration is applied by circulating a refrigerant-cooled coolant, or non-freezing liquid, through hollow electrodes or welding tips to prevent the tips from getting so hot from their contact with the sheets being welded, and from the passage of the electric current, that they fuse and stick to the sheets.

There seems to be two schools of thought regarding the value of refrigeration to the welder, one of them rating it much higher than the other. Perhaps both of them are right within their own reservations. Certainly the refrigerated welding tip is no cure-all for improper welding practices. With or without refrigeration, the surfaces to be welded must be kept cleaned; proper current and pressures must be applied; sheets and points may not be allowed to skid, and proper time intervals must be maintained. But assuming proper welding practices, there is proof of the benefits of refrigeration in welding both from laboratory tests and shop experience.

In spot welding of aluminum sheets, the "pick up" of aluminum on the copper welding tip due to fusion of the aluminum in the sheet with the copper in the tip is very materially retarded. When the tip is maintained at 70F by circulation of well water, 600 or 700 welds generally may be made before the accumulation of aluminum pickup on the tip must be cleaned off to avoid interference with efficient welding. Of course, this is a considerable improvement over no cooling at all, where only three or four welds may be possible between cleanups. But when the tip is refrigerated to 0F, upward of 2,400 welds are made before cleaning becomes advisable. Since cleaning and changing electrodes requires considerable time, an increase in number of welds between cleanups is a worthwhile saving.

Refrigeration also greatly retards mushrooming of the tip under welding pressure. In fact, this tendency is so pronounced that electrolitic copper may be used in the tips instead of the harder alloys required to prevent excessive mushrooming, if refrigeration is not employed. This improves electrical conductivity. Conductivity is further improved because electrical resistance of a conductor is reduced as its temperature falls.

A company in Detroit supplies a refrigerated welding tip that may be readily removed from the electrode, thus eliminating the removal and dressing of the entire electrode. This tip is hollow and is provided with internal fins for increasing the surface exposed to the coolant, which is circulated through the tip itself. This insures the lowest possible tip temperature for a given temperature and quantity of coolant circulated. This company also supplies a complete, self contained, cabinet type refrigerating unit for chilling and circulating the coolant.



Cut away view of one type of refrigerated electrode.

These applications of refrigeration to various types of metal working and machining shop operations by no means complete the list. However, enough has been told to indicate the efforts refrigerating engineers are making in behalf of the general war effort. Sometime there will come a peace. Along with the universal conversion of swords to plowshares, these new usages of refrigeration created to speed material for war will be applied to the wider practices of peace.

A Tug of War

IN the installation of tempering units in a fan system taking outside air at low temperatures, it must be borne in mind that the average pump vacuum is about 9 or 10 inches, whereas, in some weather, the condensation rate in the coils can reach such a value that the result may easily be a vacuum of well over 20 inches.

Under such conditions, the pump cannot remove the condensate and the coils will waterlog and, when waterlogging starts, freezing may quickly result.

The same condition, save for the possibility of freezing, may result in the case of a water heater, if the draft of water is sufficient to cause the water surrounding the heating element to become very cold. In this case, in addition to the waterlogging of the coil, severe water hammer is almost certain to result.

Due attention must be paid to the possibility—even probability—of such a condition and provision made to counteract it, by the use of adequate vacuum breakers, or other suitable means.

—John J. Woolfenden, Engineering Staff, Smith, Hinchman & Grylls, Inc., Detroit.

Radiant Heat Installed in Chapels at Naval Air Station

F. E. MARKEL
Manager, Mechanical Department, Robert & Co.

Placing coils of ¾-in. pipe before pouring the concrete floor.

INTERESTING design problems had to be solved to produce the attractive Chapel Group at the Naval Air Station, Fla. The group consists of two chapels, very similar in appearance, connected by a corridor that is open on one side and closed at the back to form offices. The corridor leads to the Meeting Room, which is part of this group.

While the chapel buildings could house the normal attendance of worshippers, seating capacity had to be provided for the occasions when special services attracted unusually large numbers. Therefore the side walls were formed mainly by sliding panels which opened up three fourths of the wall length.

By sliding back the panels, and connecting a public address system which makes it possible to hear from any part of the building, additional hundreds may enjoy the services outside of the chapel proper.

Although the area of the St. Johns River in which the station is located has a large number of sunny, warm winter days, there are a few hours at times when temperatures of 10F are reached. The maximum heating load for heating calculations was based on a temperature rise from 25F to 70F, or a rise of 45 degrees.

With the wall sections open, convected heat could not overcome the low temperatures along the floor nor be effective against the considerable movement of air through the chapel. Radiant heating was selected, and since little wall space was available for this type heating, the heating panels were placed in the floor.

Let us take a quick glance at the heating system for the Chapel Group before studying in detail the heating system for one of the two identical chapels.

Two large power stations, about one-half mile apart, supply all the steam required for heating and other purposes at the Naval Air Station. Steam is generated and distributed at 125 lb per sq in, and this pressure is reduced at each of the buildings supplied. Electrically driven condensate pumps are installed in each building to return the condensate to the power station.

For the Chapel Group, the supply mains come to the heater room below the Meeting Room which is near the center of the heating load. The meeting room, offices and minor spaces are provided with standard gravity steam radiation. Pressure is initially reduced to 40 lb for the domestic water heater, and for the instantaneous heater of the space heating system. A second valve reduces the steam to 5 lb pressure for the low pressure system. In this room are the condensate pumps, the circulating pump for the domestic hot water, and the pumps for the space heating.

The structural concrete floor of the chapels is placed directly upon the earth fill and is waterproofed against ground water. The ground water level appears to be but a few inches below the yard surface. In the Sanctuary, the floor is finished in marble tile and quarry tile is used elsewhere in the chapel.

Available information regarding heat loss through

Dedication of the Catholic Chapel at the Naval Air Station, Fla.

floors under suction conditions is not conclusive. A U-value of 0.5 Btu was used, and later performance of the system showed that the selection was proper. Estimates for heat loss included 7,000,000 Btu for ventilation, the probable excess heat from persons in the audience. Calculating for emissives:

$$\frac{\text{Total heat emission from the area}}{\text{Sq ft surface of walls, glass, ceiling}} = \frac{1{,}100{,}000}{10{,}250} =$$

108 degrees per sq ft per hour

With a factor of 90 for this surface, we have a mean radiant temperature of 54F. This is based on the formula

$$q = e\left(\frac{0.1730 \times T^4}{100{,}000{,}000}\right)$$

Where q = total radiation, Btu per sq ft per hour
e = emissivity
T = absolute temperature, degrees Fahrenheit

In establishing the basis for radiant heat, our preference for floor temperature is 80F. To provide some surplus for counteracting possible unknowns, a temperature of 100 degrees was permitted for the aisles. Heat emission by convection is as follows:

Surface Temp.	Room Temp.	Btu per Sq Ft per Hour
80	70	8
100	70	26

From Table I of Guide, and with factor of 90 for emissivity, radiant heat rates are:

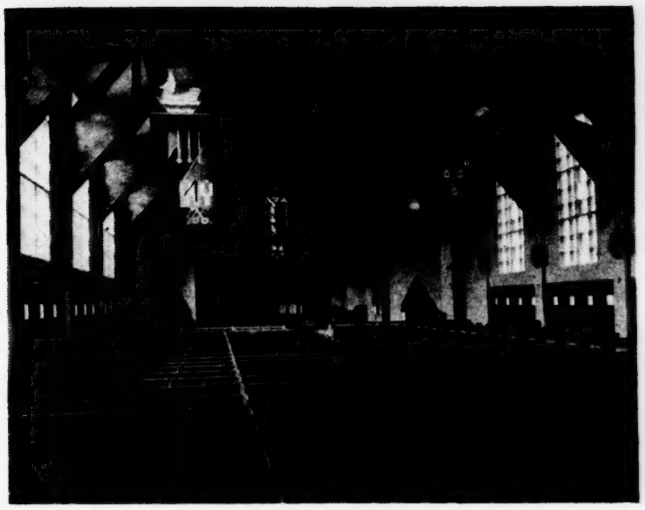
Mean radiant temp.
80 ..... 132 Btu per sq ft
100 ..... 152.6 Btu per sq ft

Applied to the areas we have the following heating values:

Floor Degree	Type Heat	Btu	
80	radiant	4400 × (132-108)	105,600
80	convected	4400 × 8	35,200
100	radiant	800 × (152.6-108)	35,680
100	convected	800 × 26	20,800
Available heat from floor			197,280

When designing the heating coils, we used the accepted 3.5 Btu per hour per sq ft of external surface of pipe. From this we evolved the following values:

Location	Sq Ft of Pipe Surface	G.P.M.	Size of Supply Pipe
Side aisle—right	66	3.96	1-in.
Side aisle—left	66	3.96	1-in.
Center aisle	88	5.28	1-in.
Seating area	508	30.50	2-in.
Narthex, baptistry and toilets	78	4.68	1-in.
Balcony	128	7.70	1-¼ in.
Altar	82	4.92	1-in.
Total	1016	61.00	3-in.



Interior of the Catholic Chapel.

The water circulating pumps of 65 gpm each are designed for a resistance of 18 ft water head. This includes 12 ft within the chapel system, and 6 ft in the remainder of the piping, heaters, valves and other parts. Temperature control was carefully considered for we had a large floor area with a possible temperature of 80F or higher, and an estimated temperature of 100F for aisles. On mild winter days, or in the intermediate season, this could produce excessive room temperatures.

It is good practice to maintain a constant water flow to preserve an even temperature over the entire panel. Temperature control is provided in a by-pass around the heat exchanges, with a three-way valve on the pump header. The valve is of modulating type, and is controlled by a modulating thermostat having the remote bulb buried in the chapel floor, one inch beneath the surface of the concrete. An outside compensating bulb further modulates the control according to outside conditions.

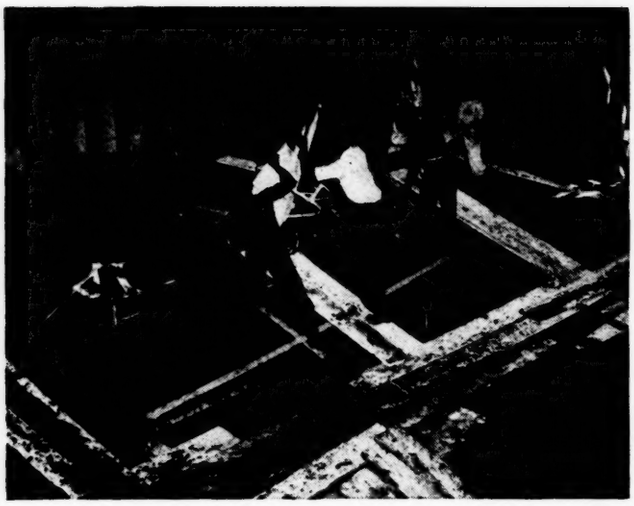
From the estimated low temperature of 25F the moderation of the weather up to 70 degrees will increase the amount of by-passed water until the floor temperature drops from 80 to 75F. As this is the minimum level required, the auxiliary switch stops the circulating pumps.

Temperature Maintained Automatically

The steam-to-water heat transfer is conducted in an instantaneous type water heater. Temperature is automatically maintained by a tandem unit consisting of pressure reducing valve and temperature regulator.

These valves are set by hand. It is planned to keep the space-heating water at a temperature close to the heating requirements so the three-way valve will not be required to provide so much control through the by-pass.

Make-up water for the space-heating system is controlled in the usual way by a tandem unit con-



View towards the Sanctuary. Placing concrete over the completed heating panel.

sisting of a pressure regulating valve and a pressure relief valve.

Instrument cost while modest, does not include any recorders. Indicating pressure gauges are placed on the low pressure side of all pressure regulating valves and on the discharge header of pumps. Indicating thermometers are set on heat exchangers and on the return header of the pumps. More instruments may be added. All heating panels in the concrete slabs, being inaccessible, are of wrought iron and all joints are welded.

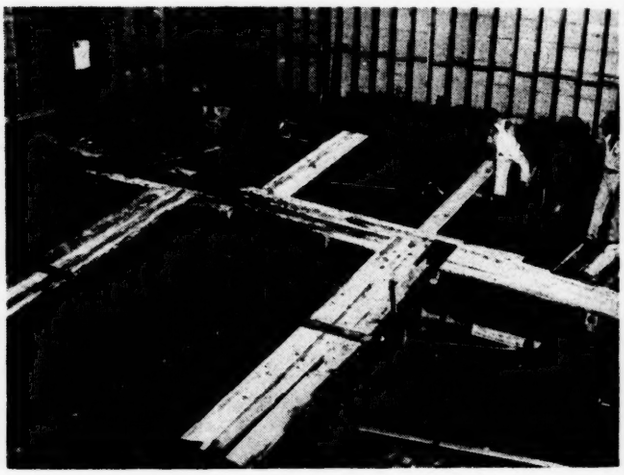
For work outside the concrete slabs, the specifications permitted threaded fittings but urged the use of welded joints. The contractor agreed; an all-welded job was secured. In general, heating panels were used as part of the mains to reduce the amount of inactive pipe.

The gallery at the rear has a panel beneath each row of seats. A riser piece forms the connection between each two levels, and at the highest point there is an expansion tank. There are additional steps between the floor levels in the area enclosed by the chancel rail. Here again, the panels are built for each step and are connected by riser pieces in the circuits.

It is only at the completion of the heating circuit that we establish a definite "main" which returns to the heater room and connects, through a valve, into the pump header.

When the chapel wall sections are closed, ventilation is necessary. Air is exhausted at low velocities by the usual equipment.

The heating plant was completed in time to be used through most of the cold weather of the past heating season. The controls work quite satisfactorily and, judging from the comments of those who use the buildings, the functioning of the plant has more than justified our selection of radiant heat for these chapels.



With the piping and reinforcing in place, the floor is now ready for the concrete.

Wood Bark for Steam Generation

Due to the war, logs in some cases are no longer being debarked in the woods but instead this is done at the mill. The bark concentrated at this one point is thus available for fuel. For example, one large paper company in the State of Maine has installed barking drums and connected refuse-burning boilers to existing steam mains. With the use of these facilities it has been conservatively estimated that there will be in this one case a fuel saving of about 7000 tons of coal per year, a saving in transportation of substantial amount for fuel within a critical area, and an insurance against a fuel shortage of any kind. The steam demand increases, of course, with increase in the production of paper, but likewise the quantity of bark available for the production of heat also increases. The resinous bark is high in fuel value and has some possibilities in the use of its ash as a fertilizer, but in any event must be disposed of either at the woods or at the mills.

In the present case a 550 hp steam boiler with forced and induced draft fans is fired from a wood-burning furnace set in a dutch-oven and equipped with a refuse storage bin with necessary feeders and chutes to the furnace. A tubular air heater is provided to supply air at the temperature necessary for the efficient combustion of the wet bark, the heat required for drying of the bark being obtained from the burning bark.—*T. W. Reynolds*

FLAWLESS

THE MAGNAFLUX TEST, shown above in Tube Turns' laboratory, quickly reveals any hidden defects in the finished product.

Flawless welding fittings are even more essential than perfect pipe. For extreme velocities, high pressures, vibration, strain, corrosive wear are *most* destructive to piping systems where flow changes direction . . . *at the fittings!* Defective fittings can cause leakage, l[illegible]kdowns, unnecessary maintenance and can disrupt [illegible]perative production schedules.

Tests to *insure* flawless fittings have been, are today and will be a permanent Tube Turns' policy.

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ARMY (E) NAVY M

TUBE-TURN

TRADE MARK

Welding Fittings and Flanges

WASHINGTON NEWS

Summarized by Loring F. Overman; Freon Control Extended to August; P. & H. Limitation Orders Relaxed; Used A. C. Program Comes to End.

While January produced no governmental action of unusual importance to the heating, ventilating and air conditioning industries, the month did develop a startling forecast:

"There will be a lot of comfort air-conditioning systems closing down next summer unless plans are made to convert to a refrigerant other than Freon-12."

This prediction, coming from F. B. Millham, administrator of conservation order M-28 (Refrigerants), carries with it the suggestion that all who can convert to methyl chloride should make immediate plans to do so. While it is realized within WPB that many air-conditioning and refrigeration systems cannot be thus converted, there appears to be no possibility that replacement charges of Freon-12 will be made available for comfort cooling systems in the event present charges are dissipated.

One reason for the continuing shortage of Freon-12 is the fact that Army's aerosol program, in which Freon is used as an expansion gas to spray insecticides wherever malarial mosquitos exist, has far outgrown original estimates. The program has been extremely successful in reducing casualties due to malaria, it is reported, and demand for the Freon-propelled insecticide has been expanded until the program is now the largest single user of the precious refrigerant.

Added to this is the tremendous demand for refrigerating equipment on naval vessels and on the mushrooming merchant marine fleet. The combination of the two—added to other war needs and to essential civilian requirements—adds up to the fact that twice as much Freon-12 is now consumed as in normal times.

New production facilities, scheduled for completion in February, but now not expected to be producing until April, will add what WPB spokesmen call, "A drop in the bucket"—approximately 1,000,000 lb per month. A second unit, scheduled for completion in September, is expected to add another million or so pounds monthly, but not until after the midsummer peak demands for cooling system replacements.

Freon Control Extended

To assure its control over all supplies of Freon-12 for war-connected needs, WPB has extended from March 31, 1944 to August 31, 1944, the restrictions provided in Conservation Order M-28. Types of business and services which are prohibited from acquiring or using any new Freon are: amusement parks; auditoriums; bank and loan associations; bowling alleys; funeral parlors; hotels and apartment houses; moving picture houses; office buildings and offices, public or private; residential buildings and dwellings of all kinds; schools; service establishments, such as laundries, cleaners and dyers, tailor shops, barber shops, beauty parlors, automobile sales and service shops, and repair shops of all kinds; studios of all kinds; animal hospitals; ballrooms, dancing studios and dance halls; bars, cocktail lounges, and beer parlors; concert halls; golf clubs, country clubs, athletic clubs, and all other clubs and club houses; night clubs; railway, streetcar and bus stations and terminals; restaurants, cafeterias, and other places selling meats, food or beverages; skating rinks; stores, selling any kind of products, material or merchandise, at retail or wholesale (excluding manufacturing establishments); and theaters.

Exemptions from these restrictions include small systems using less than 10 pounds of freon and utilized to store penicillin, blood plasma and drugs, and certain cold storage systems where a change to other refrigerants is not possible because of the construction of the system.

Heating and Ventilating Goes to Sea

While all within the heating, ventilating and air-conditioning industries are familiar with the fact that vast quantities of its equipment have gone to sea, the extent of the embarkation hinted in the U. S. Maritime Commission's annual report, submitted to Congress on January 22. A slide rule and a slight knowledge of the number of blowers, fans, ducts, condensers, compressors, heat exchangers and other units entering into the construction of cargo vessels will give some idea of the scope of the maritime equipment program.

During the fiscal year ending June 30, the report states, 81 shipyards with more than 300 ways completed 1,896 vessels aggregating 19,238,626 deadweight tons. Although construction centered mainly about the Liberty Ship, called the "workhorse of the seas", which is carrying arms, food and other supplies to fighting fronts, there was no curtailment in the C-type and tanker programs. Such ships, intended not only for war service but also to be a nucleus of the post-war Merchant Marine, are being constructed under the mandate of Congress in the Merchant Marine Act of 1936.

By the end of the fiscal year 1943 plans were virtually complete for the new Victory Ship, faster than the Liberty. The SS United Victory, the first of more than 300 contracted for, was launched January 12 at Portland, Ore. The present tentative schedule is toward vessels faster than the Liberty, and a total of 1,100 Victorys, C-types, and fast tankers are scheduled for construction this year, in addition to 800 Libertys.

Construction Expansion Vetoed

Further evidence that the industry's outlet will be military for some little time is WPB Chairman Donald Nelson's announcement, on January 12, that it has been decided to continue present restrictions on new construction and facilities without modification until the probable future course of the war becomes clearer.

WPB had been subjected to intense pressure to relax some of its restrictions on building. The decision to continue restrictions, according to Mr. Nelson, followed extended consideration by the full board of WPB.

Plumbing and Heating Division Busy

The Plumbing and Heating Branch of WPB, again doing business at its original quarters in Washington's Tempo D building—one of the only holdover temporary buildings from World War I—was busy during January revising its Limitation Orders for the purpose of reducing paper work for itself and for consumers and members of the industry.

Amendment on January 17 of Limi-

emergency:

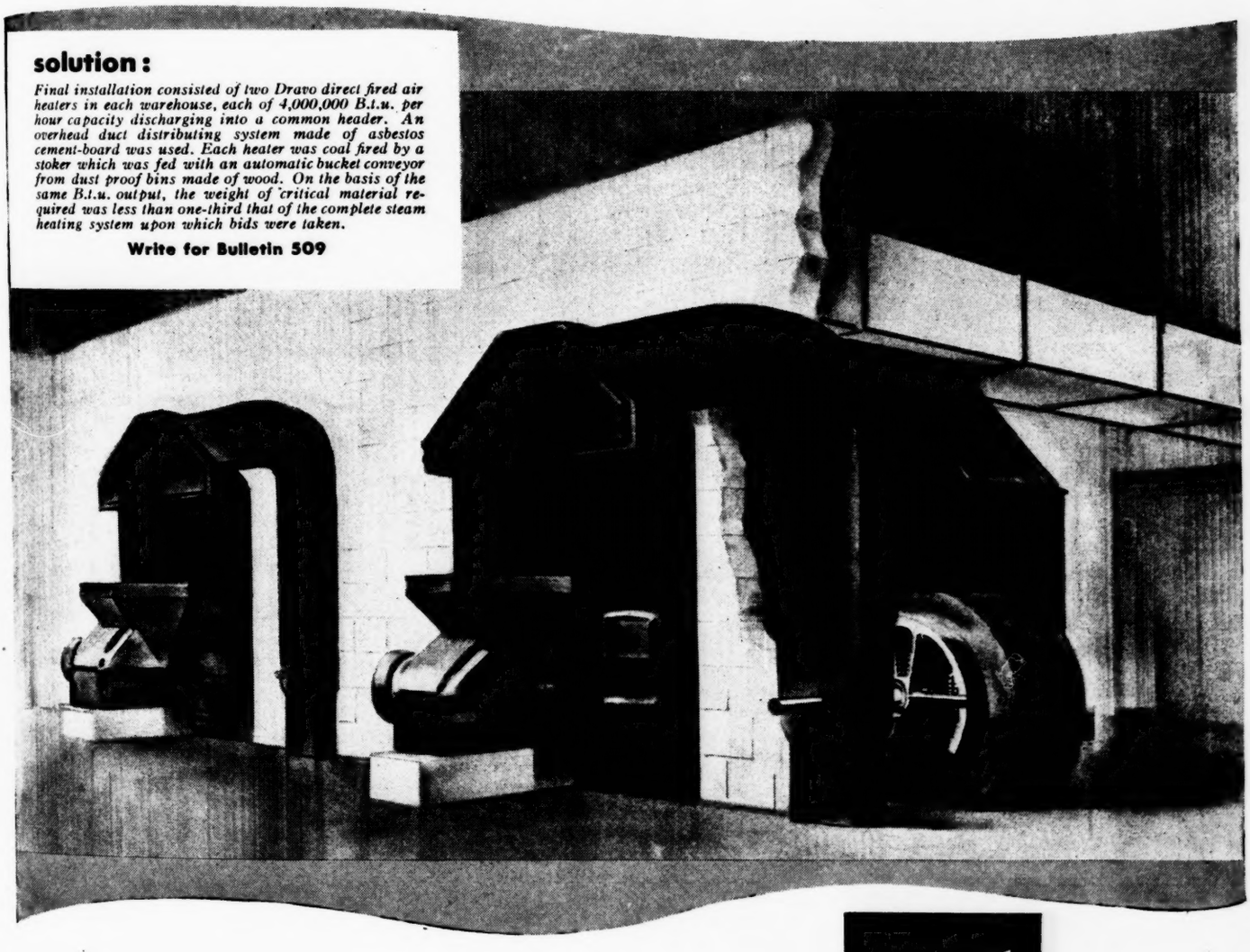
build a 32,000,000 B. t. u. heating system for 4 naval warehouses . . . but employ a minimum of critical materials

Ordinary heating problems such as economy of operation, minimum space, even heat distribution, etc., are further complicated during the present emergency by scarcity of fuel, restrictions in critical material, and manpower shortage. To heat this group of Naval warehouses, engineers were confronted with all of these problems plus the need for low cost and prompt action. Temperatures required ranged from 55° to 70° and warm air distributing duct system was involved. One of the original layouts provided for steam unit heaters fed from twin cast-iron boilers, but this system required too much critical metal; was excessively high in cost and consequently was rejected.

solution:

Final installation consisted of two Dravo direct fired air heaters in each warehouse, each of 4,000,000 B.t.u. per hour capacity discharging into a common header. An overhead duct distributing system made of asbestos cement-board was used. Each heater was coal fired by a stoker which was fed with an automatic bucket conveyor from dust proof bins made of wood. On the basis of the same B.t.u. output, the weight of critical material required was less than one-third that of the complete steam heating system upon which bids were taken.

Write for Bulletin 509



DRAVO CORPORATION · PITTSBURGH

DRAVO Heating

tation Order L-79—affecting thousands of plumbing items and a few pieces of major heating equipment—promised to lop off 7,500 consumer applications per month, and a comparable number from dealers and others in the industry. In brief, the amended order makes it possible for consumers to purchase plumbing, heating and cooking equipment, with certain exceptions, without requiring authorization from the War Production Board. Formerly an A-10 or better rating was required.

While Order L-79 is mainly concerned with smaller consumer items, relaxation of restrictions on so broad a scale in the industry is admittedly a step in the desired direction. Stokers with a capacity in excess of 60 pounds per hour, domestic type oil burners and extended surface heating equipment are specifically allocated under the Order, and furnaces, boilers, water heaters, range boilers and cast iron tubular radiators are on the restricted list.

The order provides, however, that a consumer who wishes to purchase for replacement purposes an item on the restricted list need only certify to his dealer that it is needed to replace equipment worn out, damaged beyond repair, or destroyed, and that it will not be used to replace usable equipment or to make a substitution which would provide more extensive facilities. If a consumer wishes to purchase restricted items for other than replacement purposes he must apply to his local WPB office for a preference rating.

As anticipated, amended Order L-79 permits repairmen to use an AA-3 rating to purchase such replacement items as furnaces and boilers.

Production of plumbing, heating and cooking equipment, and the use of critical materials to produce it are controlled by other orders and are not affected by changes in Order L-79.

The Heating Branch of the WPB Plumbing and Heating Division now functions with Henry Norris as chief; Lloyd B. Burlew, coal, gas and oil burning equipment section; M. C. Menacher, water heaters; L. A. Howley, cast iron boilers and radiators; M. N. Johnson, furnaces; A. Nugent, steel boilers; Carl M. Schaffer, extended surface heaters, and L. R. Moray, heating specialties and controls.

Used Air Conditioning Program Dies

The much-publicized program under which large amounts of air-conditioning and refrigeration equipment was to have been removed from civilian and government buildings and installed in war plants has gone by the board. Several efforts to induce Congress to pass a law authorizing the removal of equipment from government buildings came to naught, and it was finally decided that the cost of removing, reconditioning and reimbursing owners for used equipment was the longest way 'round.

The need for specialized types of equipment in war plants was another determining factor.

Bright Spots

Among the items mentioned in the foregoing as now available are water heaters. Both new and used direct and indirect fired water heaters may now be sold to consumers for replacement and maintenance purposes without ratings. A purchaser need only certify to his dealer that such equipment is needed to replace equipment worn out, damaged beyond repair, or destroyed, and that it will not be used to replace usable equipment or to make a substitution that would provide more extensive facilities.

The War Production Board announced January 11 it had changed specifications to permit greatly increased use of copper and aluminum strip and tubing in the manufacture of commercial and industrial coil and tube assemblies for refrigeration condensers or coolers. This action, effective January 11, was taken to make field service operations easier and to relieve producers of certain restrictions in the manufacture of coil and tube assemblies.

Relaxation of the specifications covering use of these critical metals will allow the industry an approximate increase per quarter of 300,000 pounds of copper strip, 660,000 pounds of copper tubing, and 200,000 pounds of aluminum strip. There will be a corresponding decrease of 500,000 pounds of steel strip and 400,000 pounds of steel tubing used per quarter as a result of the substitutions.

Since copper and aluminum are easier to handle than steel, manufacturers, consisting of some 25 companies, estimate that 700,000 man-hours per year will be saved, equal to 30 to 40 per cent of the entire total of industry man-hour operations.

The two amendments, which alter Schedules III and IV of Limitation Order L-126, will permit the following changes in the manufacture of coil or tube assemblies for refrigeration condensers or coolers:

1. Permit the use of non-ferrous metals in the fabrication of coil or tube assemblies for coolers of the finned type only, for coil or tube assemblies to be used in farm milk coolers of the immersion type and for pipe and tubing in water and brine coolers of the shell and tube type.

2. Remove the restrictions on the thickness of fins used in the fabrication of coil or tube assemblies for air cooling.

3. Permit the use of non-ferrous metals in the fabrication of replacement coil or tube assemblies when the assembly is to replace an existing assembly of like metals.

In addition to the relaxations permitted on the manufacturing processes, servicemen or repairmen will be permitted to use copper or copper base alloy pipe or tubing for refrigerant connections.

According to the industry's figures, there has been a reduction of 60 per cent in the number of servicemen available for field work during the past 18 months. These men are essential to keep the nation's refrigeration systems in operation and to install any new equipment authorized by WPB. Since copper tubing is much easier to use as connections, it will save considerable time if servicemen are permitted to use copper rather than iron pipe or steel tubing. Further, the use of sealed and dehydrated copper tubing manufactured for refrigerant connections will eliminate operational difficulties resulting from scale, rust, dirt and moisture usually encountered when steel tubing or iron pipe is used.

It is estimated that this change in specifications governing refrigerant connections will require an industry increase of approximately 50,000 pounds of copper per quarter.

The amended order still prohibits the use of copper or copper-base alloy pipe or tubing for service connections such as water and drain lines.

Boiler Scheduling

Scheduling jurisdiction over steam boilers of the type generally used as plumbing and heating equipment was transferred on January 22 from the Power Division of the War Production Board to the Plumbing and Heating Division. The action was taken to permit deliveries of boilers of this type to be scheduled by the Division in line with requirements for such items.

How many friends has an Air Filter?

1. ONE IS 9-YEAR-OLD BEN, whose body was racked nightly by an asthmatic cough. A device consisting of a fan and a Dust-Stop* Air Filter, installed in his window, strained out dust and pollen. Now Ben sleeps quietly for the first time in eight years.(1)

2. THOUSANDS OF factory workers also have excellent reasons for feeling friendly towards Dust-Stops. These air filters, in heating and air-conditioning systems, provide healthier and more pleasant working atmosphere.

3. HUNDREDS OF skilled technicians, finishing carefully machined parts, appreciate Dust-Stops, too. These filters protect bombsights, high-speed motors, and other delicate precision instruments which they make, from damage caused by grit, grime, and manufactured dusts.

4. AND ON MANY battle fronts, Dust-Stops can include fighting men among their friends. Why? Because Dust-Stops are contributing to the dependability of much equipment using all types of precision parts.

5. PLANT and maintenance engineers also have a friendly feeling for Dust-Stops because of the economy of these filters. As original equipment they cost only about $10.00 per 1000 CFM to install . . . less than $1.00 per 1000 CFM to replace. Find out how Dust-Stops can be friends of yours!

Owens-Corning Fiberglas Corporation, Toledo 1, Ohio. In Canada, Fiberglas Canada, Ltd., Oshawa, Ontario.

ARMY E NAVY

Ask your regular supplier about

FIBERGLAS*

DUSTOP

*T. M. Reg. U. S. Pat. Off.

*AIR FILTERS

(1) Name and full particulars on request.

NEWS OF THE MONTH

Anthracite Industries Gears Program to War

NEW YORK—Studies of the further application of barley-bituminous mixtures in industrial heating and power plants, exhaustive tests of magazine-type space heaters, distribution of a consumer leaflet on conservation, instruction of public and private building custodians in proper firing methods, and co-operation with the armed forces in similar educational work, are among the current activities of Anthracite Industries, Inc., research and educational organization of the hard coal industry, according to Frank W. Earnest, Jr., president.

Increased production of the domestic sizes of anthracite during the past two years has resulted in a temporary surplus of No. 3 buckwheat, or barley, as only a limited number of power plants have stokers or grates which can burn that size, Mr. Earnest stated. For that reason, and also because wartime fuel and transportation problems are forcing many power plants now using bituminous coal exclusively to consider mixing it with anthracite, engineers of the Anthracite Industries' laboratory at Primos, Pa., and members of the field force, have been conducting studies and experiments designed to extend the field of application of barley-bituminous mixtures. Many proportions, ranging up to 80% barley and 20% bituminous, have been tried and found to give excellent results in underfed bituminous stokers. As the percentage of anthracite varies with the conditions encountered, each barley-bituminous application must be individually tailored to obtain maximum results.

In conjunction with the U. S. Army and the Bureau of Mines, an extensive series of boiler tests with hard-soft coal mixtures have been run at an eastern army camp. Results have been so satisfactory to date, that the Army has ordered 7,500 tons of barley for use in the camp's power plant.

Refrigeration Code for Los Angeles

LOS ANGELES—A refrigeration code for Los Angeles which has been submitted to the Board of Building & Safety for consideration provides for special permits for installing, moving or altering refrigeration equipment, inspection of such work by competent refrigeration inspectors, and specifies the locations in which equipment and refrigeration rooms may be established.

Under the code, a permit would have to be obtained from the Building & Safety Department for installing or relocating a refrigeration plant or moving it wholly or in part to any other location in the same building; for changing the type of refrigerant used; or for making alterations or replacements costing more than $100 in materials or labor. Normal repair and maintenance operations would require no permit.

The code provides that all refrigeration work be inspected by a qualified inspector, with special provisions applying to emergency installations or replacement work in occupied buildings.

NAFM Elects Morse, Frank

DETROIT—At the twenty-seventh annual meeting of the National Association of Fan Manufacturers held here January 19, the following officers were elected for the ensuing year: C. T. Morse, president; J. M. Frank, vice-president; and L. O. Monroe, secretary-treasurer.

Foote Talks on Plane Heating

BUFFALO—Comfortization of Airplanes was the topic discussed by Avery G. Foote, research engineer, Curtiss-Wright Corporation, before the January meeting of the local chapter of the ASHVE here.

S. J. O'Brien Elected President

NEW YORK — S. J. O'Brien, president, S. J. O'Brien Sales Corporation, was elected president of the Electrical and Gas Association of New York at its annual meeting. Other officers elected are: First vice-president, T. F. Barton, vice-president, General Electric Company; second vice-president, T. H. Joseph, president, E-J Electric Installation Co.; treasurer, James A. Sackett, sales manager, Kings County Lighting Company; secretary, C. A. Norton, sales manager, Pierce Laboratory, Inc., and assistant secretary, John W. Walter, manager, Eastern Division, Apex Rotarex Corporation.

Gas Conference to be Held in March

NEW YORK—Plans are being completed for the War Conference on Industrial and Commercial Gas, sponsored by the American Gas Association, to be held at the Hotel Seneca, Rochester, N. Y., March 30 and 31.

The three-fold objective of the conference is to study equipment and practices in gas utilization in war production plants, military camps, and bases; to study the requirements of altering or replacing old commercial gas equipment in war plants, and to rehabilitate commercial equipment during the coming conversion period; and to plan for maximum industrial and commercial gas and equipment sales in the period after the war. This conference alternates yearly between natural and manufactured gas territories.

Post-War Manufacturing Activities

PHILADELPHIA, PA.—A study of post-war preparation activities of 350 major companies, including 15 in the heating, air conditioning, and plumbing fields, was made by research department of Curtis Publishing Company.

Of the companies interviewed, 58% plan to bring out some new product. New products are planned by 60% reporting in the heating, air conditioning and plumbing fields.

New products of 54% will be in the same field as the normal peacetime production while 12% plan to go into new fields, and 20% reported new products in both their own and another field. For example, a building material company will make an air conditioner.

The greatest progress in organization for post-war activities was reported by five groups, including manufacturers of heating, air conditioning and plumbing equipment.

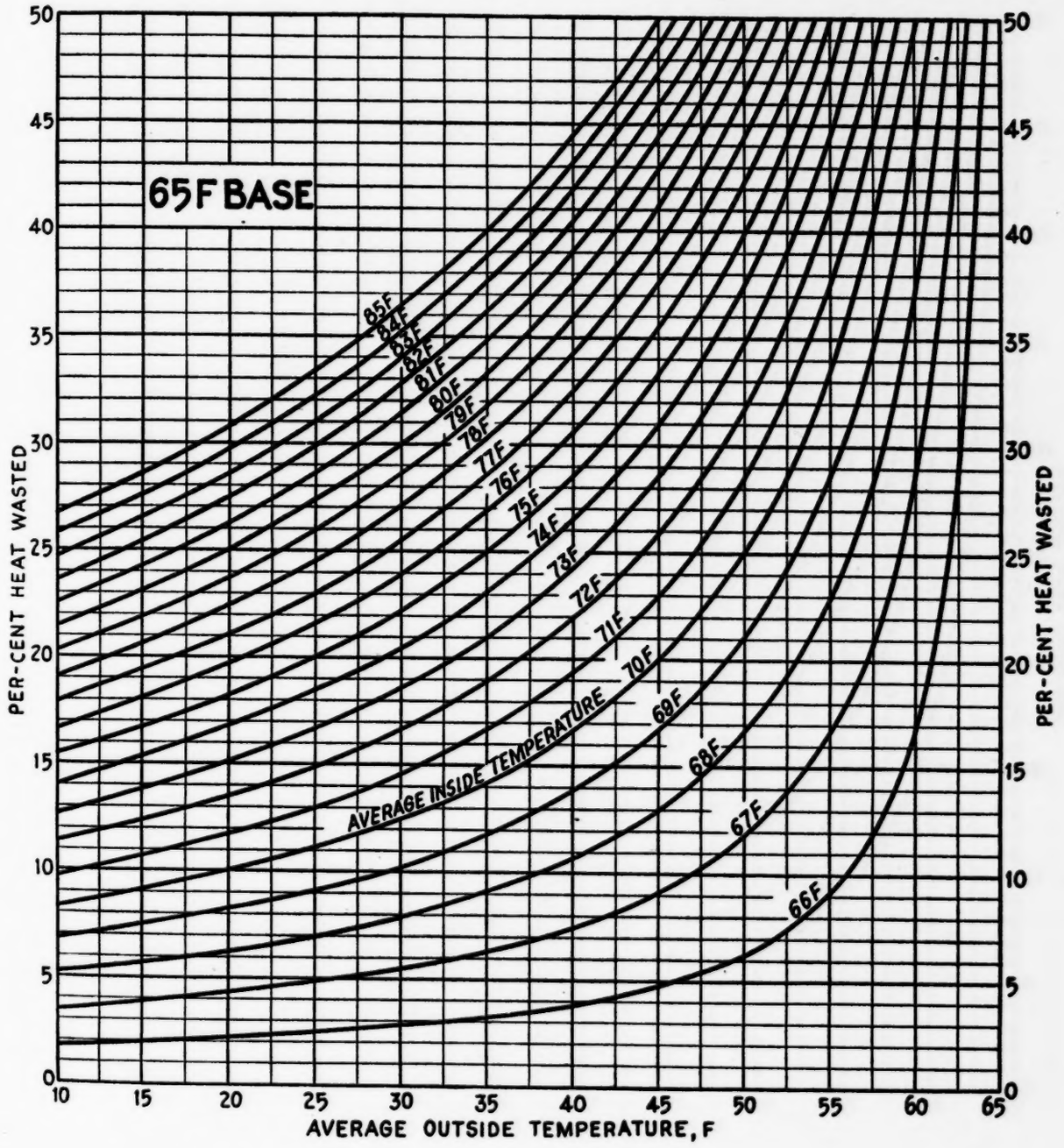
Wisconsin Oil Heat Elects Schroeder

MILWAUKEE—At the annual meeting of the Wisconsin Oil-Heat Association Walter J. Schroeder was elected president for his second term, and articles of organization to incorporate the Association were filed with the Secretary of State. The Wisconsin Oil-Heat Association was organized in March, 1922. Its membership comprises oil burner manufacturers, dealers and distributors, accessory manufacturers and distributors, oil companies and oil burner service companies throughout Wisconsin. In May, 1943 it became affiliated with the Oil-Heat Institute of America, Inc., Distribution Division and has been active in that group ever since.

Other officers elected at the annual meeting were Howard H. Elder, vice-president (Cities Service Oil Company); John Weiler, treasurer (Milwaukee Welded Products Inc.); directors: Charles N. Smith (Callaway Fuel Co.), E. R. Benson (Oil Heating Sales Company), J. H. Shellenberger (Shellenberger-Gregg Co.), and E. J. Galdabini (Perfex Corp.).

HEAT WASTED BY HIGH INSIDE TEMPERATURES

(Where 24 hr. average inside temperature is normally 65F)



Note that the "Per-Cent Heat Wasted" is in per-cent of the load under the excessive condition. To find how much more heat (or fuel) is being used than would be if 65F were carried, see other side of this sheet

Published by HEATING AND VENTILATING, *148 Lafayette St., New York.* *2/44* *Copyright, 1944, by* THE INDUSTRIAL PRESS.

HEAT WASTED BY HIGH INSIDE TEMPERATURES

(Where 24 hr. average inside temperature is normally 65F)

The chart on the reverse side of this sheet shows the percentage of heat wasted by carrying excessive inside temperatures. It can be used for either daily or seasonal calculations. Since the chart is based on a 65F temperature, it is usable for most installations where the 65F degree day applies—that is, where the 24 hour inside temperature average is approximately 65F.

Example: An apartment normally heated to 70F during the day and evening hours but with a 24 hour average inside temperature of 65F, is being heated to an average inside temperature (24 hr.) of 75F. The temperature outside averages 50F on the day in question. What per cent of the heat is being wasted? Solution: Locate 50F on the lower scale, move vertically to the curved 75F line, thence horizontally to read 40%. That is, on this day, 40% of the heat is wasted.

To use the chart for a seasonal calculation, refer to Reference Data 255-256 for average outside temperatures during the heating season for the locality in question.

Example: An apartment in Parkersburg, W. Va., normally heated to an average inside temperature of 65F for 24 hours, is being operated with an average inside temperature (24 hr.) of 72F. How much of the heat is wasted due to too high inside temperature? Solution: Referring to Reference Data 255-256, the average heating season outside temperature for Parkersburg is 45F. Locate 45F on the lower scale, move vertically to the 72F curved line, thence horizontally to read that 26% of the heat is being wasted.

Note that the answer is in per cent of the heat load under the excessive condition. In the foregoing example 26% of the heat is wasted; this does not mean that 26% more heat is being used than necessary.

The excess heat $= \frac{X}{100-X}$, where X is the percent of heat being wasted (from the chart). In this example, then, $\frac{26}{100-26} = 35.1\%$ more heat is being used than would normally be used.

Note also that the chart only applies to buildings whose average 24 hour inside temperature is 65F. The chart is not applicable unless the 24 hr. average inside temperature is known.

Reference Data 265-266 covers buildings where the 24 hr. average inside temperature is 55F.

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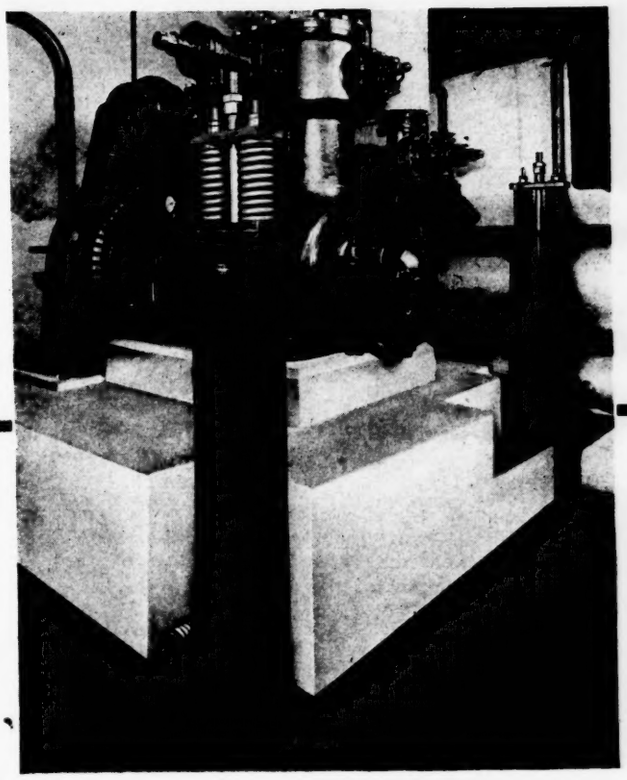
In New York's famous Center Theater

KORFUND EQUIPMENT ISOLATES VIBRATION

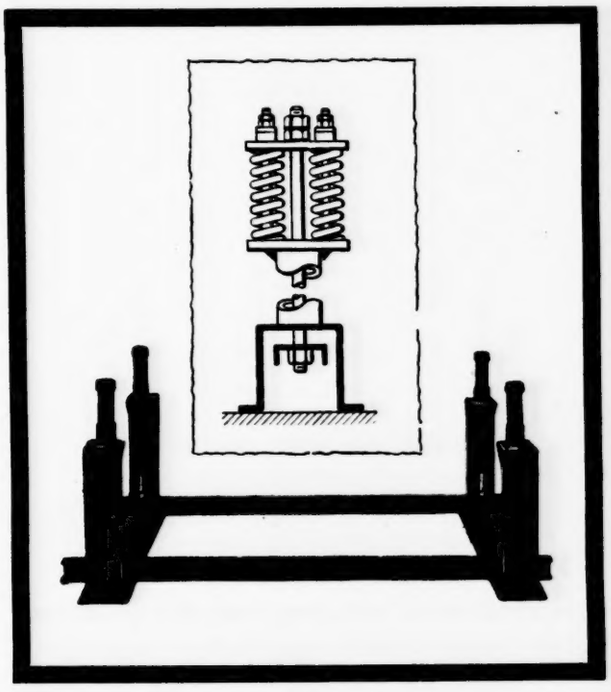
. . . assures smooth, quiet operation of Refrigeration Machinery for mammoth Ice Rink on Stage



In the Center Theater's refrigeration plant, Korfund Vibro-Isolators prevent completely the transmission of vibration. Diagram (below) shows how load is transmitted through vertical rods on which machinery is mounted, and carried on springs situated above machinery's center of gravity.

THE huge skating rink on which the "Stars on Ice" show thrills thousands daily requires continuous and unfailing operation of heavy refrigeration machinery. Synonymous with this requirement is the need for preventing the transmission of annoying vibration to any part of the theater. Korfund Vibration Control equipment (Vibro-Isolators type SUP) performed so satisfactorily on the first installation that a second unit was similarly installed.

In the photo and diagram illustrating one of the Center Theater installations, note how the Frick compressor unit and its foundation are suspended on steel springs, the load being transmitted by vertical rods fastened to horizontal steel members incorporated in the floating concrete block. The suspension arrangement permits control of vibratory forces acting in any direction. Maximum stability is assured by the fact that the SUP Vibro-Isolator arrangement enables the steel spring assembly to be placed as near in line as possible with the center of gravity of the suspended equipment. Undesirable oscillation is checked by special vertical and horizontal snubbers mounted near base of stanchion. The SUP Vibro-Isolator often eliminates the need for heavy concrete inertia bases or costly foundations, and is thus well suited for the isolation of equipment installed on existing floors.



Korfund Vibration Control is the result of 40 years of scientific research and experience gained from over a quarter of a million successful installations like the above. Whatever your vibration problem—let Korfund help you solve it as we have helped thousands of others!

Send for useful engineering data on vibration control for compressors, pumps, fans and other types of machinery.

The KORFUND *Company, Inc.*

KORFUND VIBRATION CONTROL

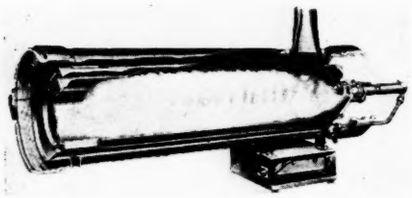
ENGINEERS • MANUFACTURERS

LONG ISLAND CITY 1, N. Y.

NEWS OF EQUIPMENT AND MATERIALS

Janitrol Aircraft Heater

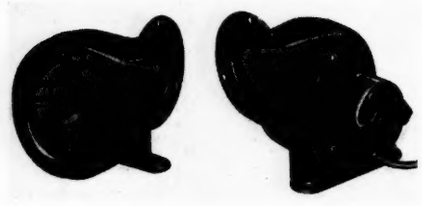
NAME—Janitrol aircraft heater.
PURPOSE—For heating airplanes especially at high altitudes. Used to defog, de-ice and anti-ice windshields, gunners' windows and turrets, to prevent ice from forming on wings and empennage surfaces, to keep bomb-



sights clear, instrument gun breaches and other equipment operative.
FEATURES—Device has whirling flame feature credited with being responsible for the unit's ability for being able to stay lit under extreme conditions such as a 700 mph dive. Said to be the only heater which has stayed lit at 37,000 ft altitude. Whirling flame gives continuous re-ignition and a self-piloting effect. Can be lighted at —70F.
SIZES AND CAPACITIES—Available in four sizes with range from 15,000 to 125,000 Btu per hr. The larger size runs one hour on slightly more than a gallon of gas.
MADE BY—*Surface Combustion, 2511 Terminal Tower, Cleveland, Ohio.*

Ilg Utility Blower

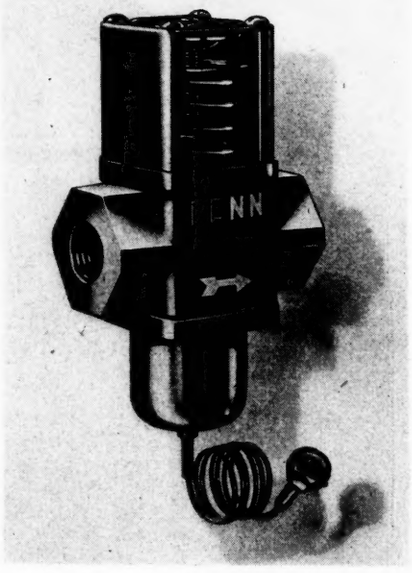
NAME—Ilg 6S utility blower.
PURPOSE—An all-purpose blower developed especially for applications on machinery.
FEATURES — Intended especially for



machines with built-in air handling equipment, this blower is said to make available a wide variety of special mountings on a mass-production line of blowers, giving the user a unit that is practically custom-built for each application. Eight different arrangements are available which include blower only; blower and stand only; blower and inlet flange only; blower and discharge flange only; blower, stand and inlet flange only; blower, stand and discharge stand only; or blower, stand, inlet and discharge flanges. Housing and stand of die-stamped steel, multi-blade wheel, zinc die-cast. Powered by series-wound, 110 volt, single phase, 60 cycle sleeve-bearing-type AC motor that operates at 3400 rpm. A length of cord is brought out of the motor for connections.
LITERATURE AVAILABLE—Bulletin containing performance curves, dimensions and operating data.
MADE BY — *Ilg Electric Ventilating Co., 2850 No. Crawford Ave., Chicago 41, Ill.*

Penn Water Regulating Valves

NAME — Penn series 246 commercial and series 246-N Navy type water regulating valves.



PURPOSE—For water cooled refrigeration compressors and condensers, stationary and marine service.
FEATURES—Said to incorporate great sensitivity to refrigerant head pressure, minimum spring power requirement, and insure free movement of all working parts. New and inherently different design claimed to eliminate sticking of seats, water hammer, drain plugs, rusting of range springs, need for lubrication, corrosion of, and sedimentation on sliding parts. Inlet and outlet water pressure forces on the port are controlled by use of two rubber diaphragms which also serve as gaskets to prevent leakage. Range spring and sliding parts are not submerged in water. Use of rubber diaphragms and internal design of valve body eliminate necessity for drain plug. Drainage of the valve is not necessary for winter or other standby periods because if freezing occurs, the rubber diaphragms compensate for any expansion with the valve body.
SIZES AND CAPACITIES—Available with threaded or flange type connections in sizes from ¼ to 4 in.
LITERATURE AVAILABLE — Bulletin R-1986.
MADE BY—*Penn Electric Switch Co., Goshen, Ind.*

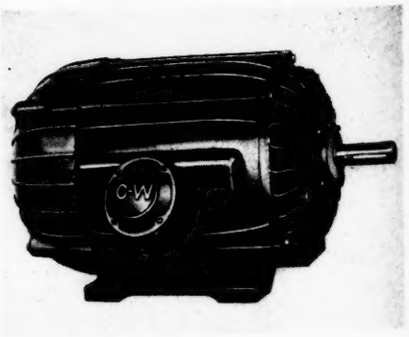
Speed-Ak Templates

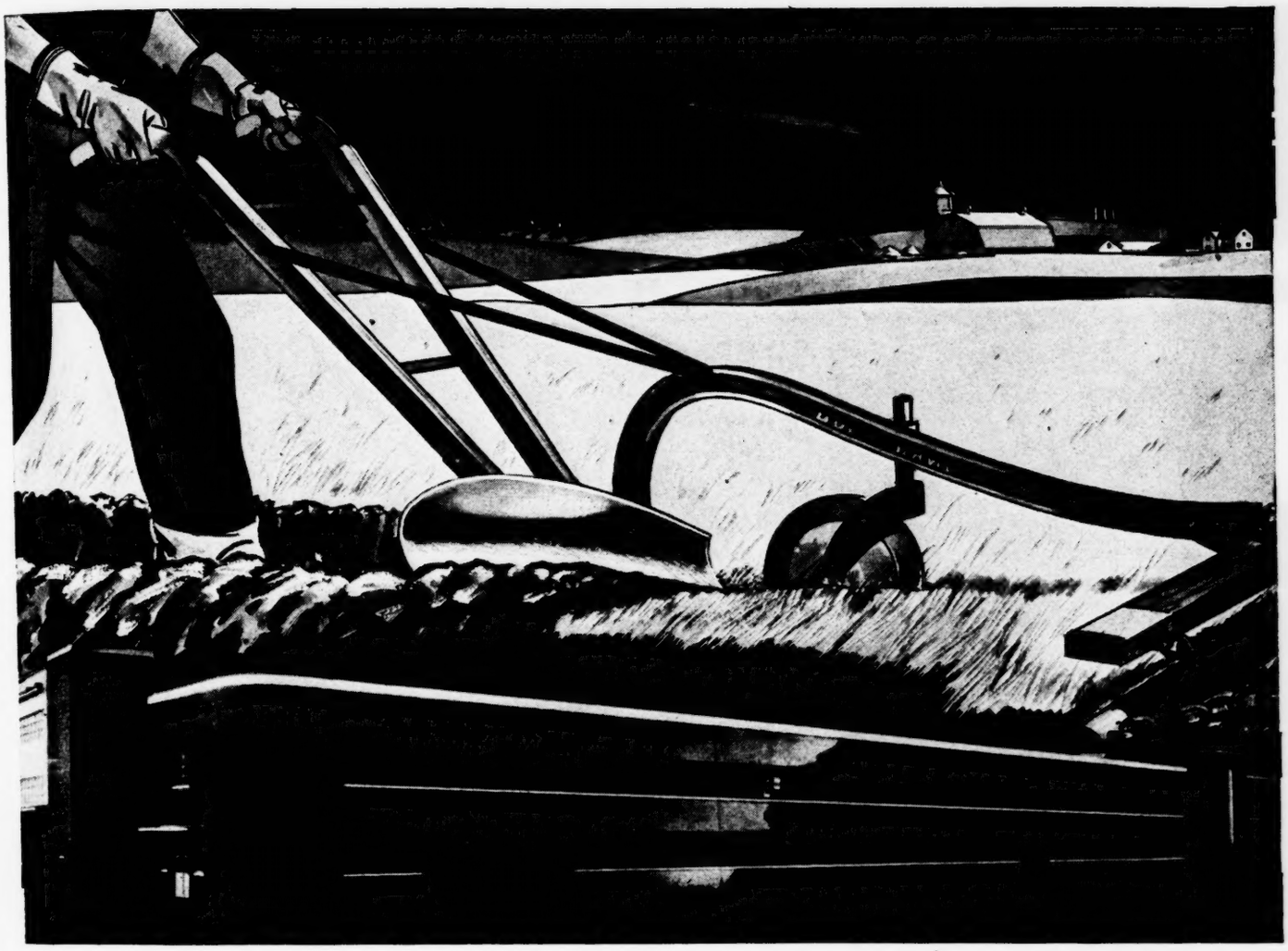
NAME—Speed-Ak templates
PURPOSE—For laying out steel pipe to indicate cuts in forming ells, tees, Y's, etc.
FEATURES—Speed-Ak stretch-cuts are accurate, inexpensive, and durable templates which may be wrapped around the pipe. They are also useful for laying out connections to be formed from flat sheet metal. Complete sets are available for all standard pipes up to 12 in. O.D. Maker reports that with these templates a welder can lay out his cuts in one-tenth of the time required for plotting the outline of the pipe.
MADE BY — *Engineering Associates, P.O. Box 1431, Fort Worth 1, Texas.*

Corrosion-Resistant Motor

NAME—Sealedpower motor.
PURPOSE — For operation in atmospheres containing injurious dusts, corrosive vapors, and gases.
FEATURES—Motor is of the totally en-

(Continued on page 94)





More than a quarter million dollars plowed back to American industry

by One Allen Ventilating Unit . . .

Ventilation that does the job comes first with Allen. Close on the heels of that consideration comes lowered cost to the buyer. Only when we click on both are we satisfied, because only then do we know we've done the right kind of job.

Basically better ventilation engineering has enabled Allen to furnish industry with more than four thousand Allen Type "H" Roof Fans at an average saving of $62.50 per unit over the nearest comparable equipment—more than a quarter million dollars.

The explanation is simple. Allen patents cover the use of dampers balanced in the effluent air stream—open when the fan is running, closed automatically when power is off. This is in contrast to construction used by several other manufacturers wherein mechanical, motor-operated dampers are mandatory and up the cost.

Approximately 400 of these Allen units top the huge Willow Run Bomber Plant, shown above. Allen has an answer for your ventilation problem, too, whether it's ordinary or extraordinary. Let's talk it over. *The Allen Corporation, 9751 Erwin Avenue, Detroit 13, Michigan.*

SOME TYPICAL USERS OF ALLEN TYPE "H" UNITS

BLACK AND DECKER
BOWER ROLLER BEARING
BUNDY TUBING COMPANY
EX-CELL-O CORPORATION
FISHER BODY DIVISION
FORD MOTOR COMPANY
GENERAL ELECTRIC COMPANY
DES MOINES ORDNANCE PLANT
TWIN CITIES ORDNANCE PLANT
LOWELL ORDNANCE PLANT
ACME COTTON PRODUCTS
KEN-RAD CORPORATION
BROOKLYN NAVY YARD
MINNEAPOLIS ORDNANCE PLANT
SERVEL, INCORPORATED
A. C. SPARK PLUG
GAR WOOD INDUSTRIES

THE *Allen* CORPORATION

ENGINEERED VENTILATION FOR INDUSTRY

closed, fan cooled type with no cooling ducts to become fouled. Each coil is individually sealed against moisture, fumes and dust.

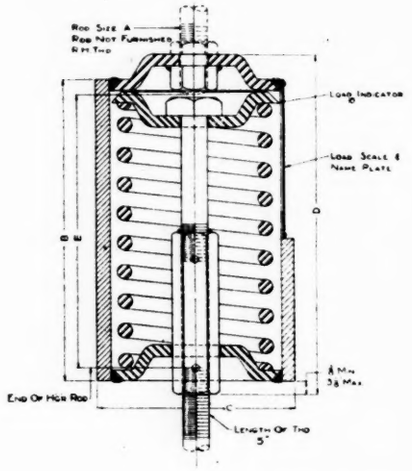
SIZES AND CAPACITIES — 1 to 15 hp, and for operation with any polyphase power supply.

MADE BY — *Crocker-Wheeler Division of the Joshua Hendy Iron Works, Ampere, N. J.*

Pipe Hanger

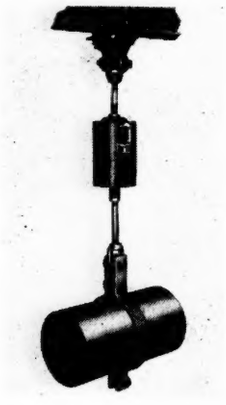
NAME—Grinnell pre-engineered spring hangers.

PURPOSE—To minimize changes in the supporting force of the hanger



during change in the vertical action of the piping.

FEATURES—Designed to meet requirements of A.S.A. code for pressure



piping. Problem of erection is simplified by device having a scale plate on side of each hanger where the deflection and load is indicated, insuring correct installation. A coupling eliminates necessity for a turnbuckle to pick up load.

SIZES AND CAPACITIES—14 sizes with a load range from 84 to 3200 lb. Each size permits uniform variation in supporing force of not more than 12½% in ½ in. deflection of the spring.

LITERATURE AVAILABLE — Four-page bulletin "Grinnell Pre-Engineered Spring Hangers."

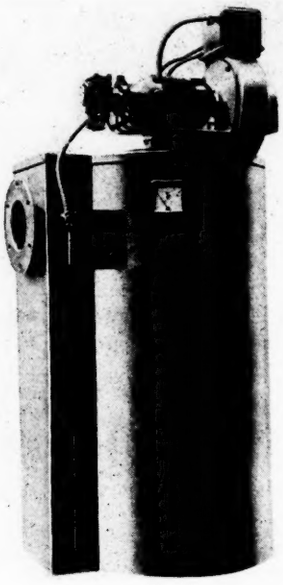
MADE BY — *Grinnell Company, Inc., Providence, R. I.*

Hot Water Generators

NAME—Agitair oil burning hot water generator.

PURPOSE—For heating and hot water service.

FEATURES — Weight and cost of installation have been reduced by simplifying many of the usual accessories. Operation is fully automatic. Diesel or lighter fuel oil is



burned under positive pressure supplied by a low speed motor.

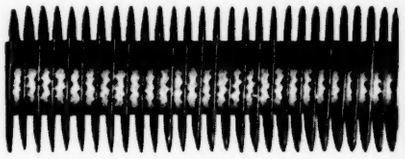
MADE BY — *Air Devices, Inc., 17 E. 42nd St., New York 17, N. Y.*

Lotemp Finned Pipe

NAME—Industrial Lotemp finned pipe.

PURPOSE — Heat transfer surface for low temperature industrial applications.

FEATURES—Manufactured in standard pipe sizes, furnished in straight lengths, for use in the fabrication of refrigeration equipment. A continuous steel ribbon is wound spirally on the pipe; the method of attaching is said to result in a positive and permanent bond between fin and pipe. Assembly hot dipped in zinc bath for resistance to corrosion and permanence. Adaptable to a wide range of refrigeration uses especially adapt-



able for low temperature applications.

SIZES—Available in a variety of pipe diameters, fin widths and fin spacings.

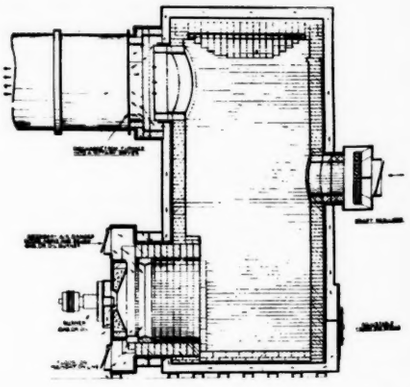
MADE BY — *Industrial Manufacturing and Engineering Co., 3845 N. Ravenswood Ave., Chicago 13, Ill.*

Hot Gas Generating Furnace

NAME—Agitair hot gas generating furnace.

PURPOSE—To produce hot gas at any desired temperature, for use where material being processed is not affected by contact with the gases.

FEATURES—Constant discharge temperatures are maintained by modulated controls and a reversing type burner control motor. Furnace will operate down to 25% of full capacity. Hot gases said to be without smoke or odor except that due to sulfur in the fuel. Adjustable dampers regulate fresh air intake and draft in firebox.



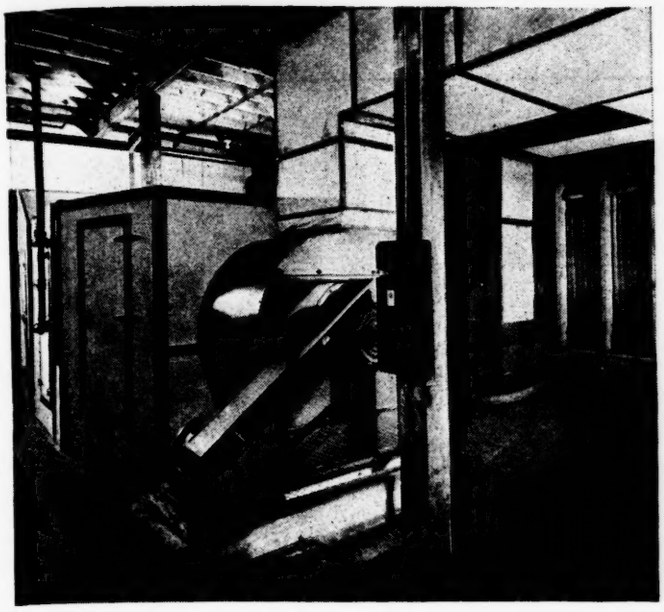
MADE BY — *Air Devices, Inc., 17 E. 42nd St., New York 17, N. Y.*

Slide Rules

NAME—Slide rules.

PURPOSE—To meet increased demand for this type instrument three new slide rules are available, ane for professional draftsmen, one for appren-

(Concluded on page 96)



Carey FIREFOIL PANEL

Saves Vital Metal . . .

Gives Top Efficiency in

AIR-CONDITIONING SYSTEM at

Chrysler Corporation's Dodge Chicago Plant

In planning Chrysler Corporation's Dodge Chicago plant, top efficiency was the keynote throughout—in the building itself, its equipment, and construction materials used. For some of the air-conditioning system, CAREY FIREFOIL PANEL was selected for the same reasons it has been specified for other important installations—its inherent advantages, plus savings in war-vital metal.

CAREY FIREFOIL PANEL is a unique structural insulator. It is a hardened asbestos insulating board, consisting of corrugated asbestos felts firmly bonded and surfaced on each side with asbestos-cement sheathing. The result is a strong, rigid, fireproof, sound-deadening material. Eliminates vibration, is light in weight, and waterproof—will not rust and requires no painting.

CAREY FIREFOIL PANEL is designed especially for constructing plenum chambers, large ducts, and housings for air conditioning and other equipment. For information, write Dept. 49.

Illustration shows use of Firefoil Panel in air-conditioning unit installed by Narowetz Heating & Ventilating Co., Chicago, Ill. Plant designed by Albert Kahn, Associated Architects & Engineers, Inc., Detroit, Mich.

THE PHILIP CAREY MFG. COMPANY • Lockland, Cincinnati, Ohio
Dependable Products Since 1873
IN CANADA THE PHILIP CAREY COMPANY LTD. Office and Factory, LENNOXVILLE P. Q.

PREVENT

Rust • Scale • Dirt

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"The Screen's the Thing"

- ***A High Grade Monel Woven Wire Basket*** with mesh fine enough to catch the dirt — yet ample free area for passage of clean condensate, oil and other fluids.
- ***Body Cadmium Plated Inside and Out for Protection Against Corrosion.***
- ***Readily Removed Steel Blow-Off Bushing.***
- ***Bushing Automatically Aligns Screen.***
- ***Thousands in Use—Sold by Over 100 Mill Supply Houses.***
- ***6 Sizes ½" to 2" for Pressures up to 600 Lbs.***

See Your Supply House or write for Bulletin S-200

YARNALL-WARING COMPANY

104 MERMAID AVENUE • PHILADELPHIA 18, PA.

YARWAY STRAINERS

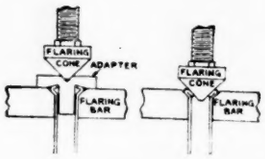
tice draftsmen, and also a 5-in. pocket type slide rule. With the exception of the apprentice type, which has a painted face, all rules have celluloid faces.

MADE BY—*The Frederick Post Company, Box 803, Chicago 90, Ill.*

Flaring Tool for Plastic Tubing

NAME—Imperial multi-size Double-Flaring tool.

PURPOSE—Developed specifically for flaring plastic tubing.



FEATURES—Flare made by this tool is known as "double-flare" because plastic tubing is folded back at the ends to form a flare with double thick walls. Such flares are said to retain their shape after flaring. The double thickness protects

against the wall being squeezed too thin and the joints have great resistance to pull-out.

SIZES AND CAPACITIES — Available for ⅜, ½, ⅝ and ¾ in. outside diameter tubing with .062 in. wall. Complete tool consists of steel flaring bar, forged steel yoke with swivel cone and four adapters, complete in metal kit.

MADE BY—*The Imperial Brass Mfg. Co., Chicago 7, Ill.*

CS_2 Analyzer

NAME—Ultraviolet photometer.

PURPOSE—Gas and vapor analyzer for determining dangerous concentrations of carbon disulfide in the atmosphere. Analyzer can take quick samples or run continuous samples and give true and instantaneous readings. This permits accurate second-by-second observation level in each step of a manufacturing process. Operation of the device is based on the phenomenon of light absorption by gases. Most gases absorb light of some particular wave length, in effect casting a shadow where that particular wave length light would otherwise have fallen. In a spectrum this shadow is an absorption line. Analyzer is so constructed that the air to be analyzed is pumped through several small chambers, which filter

out dust, oil and moisture, and thence into a pair of parallel tubes, about 31 inches long. The contaminated air runs into the first tube and then through a canister of activated charcoal which removes the carbon disulfide, and passes purified air into the second tube. In this way there may be a continuous comparison of the purified with the contaminated air and very minute differences may be detected.

DEVELOPED BY—V. F. Hanson, of the Electrochemicals Department of the E. I. DuPont de Nemours and Company (Inc.) designed the original instrument, and a modified model, intended particularly for carbon disulfide analysis, has been developed by Dr. Shirleigh Silverman, assisted by Dr. J. W. Ballou and W. H. Warhus, all of the Rayon Technical Division of the company. The Mine Safety Appliance Company, Pittsburgh, is planning to manufacture instruments of this general type.

Positioner for Motor Valves

NAME—Foxboro Vernier Valvactor.

PURPOSE—To insure precise and dependable valve response where operating controls may be slow or uncertain.

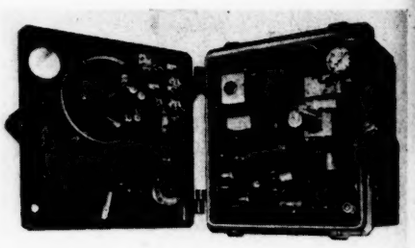
FEATURES — While the new unit retains the outstanding advantage of relay action, new type of bearing is used in place of pivots to eliminate mechanical vibration. This type of unit has had eight years of widespread use.

MADE BY — *The Foxboro Company, Foxboro, Mass.*

Pneumatic Indicating Controller

NAME—Tag Celectray.

PURPOSE — Electronic instrument for



controlling changes in air pressure.

FEATURES—As temperature varies, a light image moves across a controlling edge of a flag and thus varies the amount of light that may reach a phototube. More light on the tube causes a greater current flow which raises the controlled air pressure. Less light on the tube lowers the controlled air pressure.

MADE BY—*C. J. Tagliabue Manufacturing Co., 550 Park Ave., Brooklyn 5, N. Y.*

Electric Unit Heaters

NAME—Electromode industrial electric air heaters.

PURPOSE—For heating isolated buildings or spaces where intermittent heating is desired.

FEATURES—Two new models of heaters have been placed on the market.

SIZES AND CAPACITIES—In either portable or suspension models, capacities from 5112 to 204,900 Btu per hr.

LITERATURE AVAILABLE—Bulletin 65.

MADE BY — *Electric Air Heater Co., Division of American Foundry Equipment Co., Mishawaka, Ind.*

BUILD A PERMANENT PROFITABLE STOKER BUSINESS *with...*

FREEMAN STOKERS

FIRST CHOICE OF AMERICA'S FOREMOST HEATING ENGINEERS

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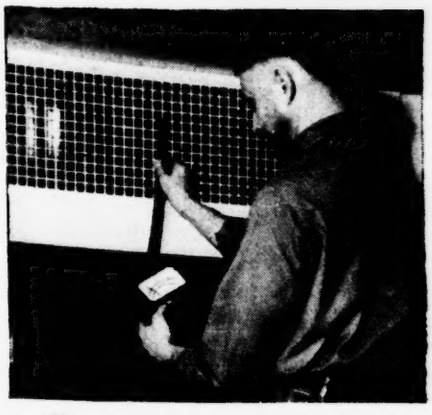
PROMPT DELIVERY

Complete Range of Sizes: 65 to 1200 Pounds Per Hour

Freeman offers you an outstanding opportunity to render vitally important wartime service in cooperation with the government's fuel conservation program . . . *and, at the same time, to develop your own permanent, profitable stoker business!* Don't wait; get complete details now. Freeman stokers have proved their superior efficiency and economy in thousands of installations throughout the world . . . are ruggedly constructed for heavy duty and long life . . . are made and guaranteed by a pioneer stoker manufacturer, in business since 1864 and one of the industry's leaders in national sales volume! Inquire about your territory today:

Alnor VELOMETER



★ all purpose air velocity meter

Instantaneous, direct readings of air speed measured in feet per minute with the *Alnor* Velometer give you the quick, accurate information on air movement needed to check operation of blowers, fans, air conditioning installations, and similar equipment. No calculations, no timing, no conversion tables; read velocities direct from the Velometer scale. Extension jets permit accurate readings in many locations that would be completely inaccessible with other means of measurement.

The Velometer is made in several standard ranges from 20 fpm to 6,000 fpm, and up to 3 inches static or total pressure. Special ranges available as low as 10 fpm and up to 25,000 fpm velocity and 20 inches pressure.

Write for Bulletin No. 2448-E.

Illinois Testing Laboratories, Inc.

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DE-STA-CO Blower Wheel Housings

of Standard Dimensions

Made in sizes to fit 4½, 5, 6, 7½ and 9 inch wheels and in widths to suit single or double inlet wheels . . . also individual parts for your own assembly.

De-Sta-Co housings are engineered for maximum efficiency and quietness in operation . . . neatly finished. Send for "De-Sta-Co Blower Housings" bulletin; gives complete information, including essential dimensions.

DETROIT STAMPING CO.

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Degree-Days for December, 1943

HEATING AND VENTILATING'S 16th year of publication of monthly degree-day data

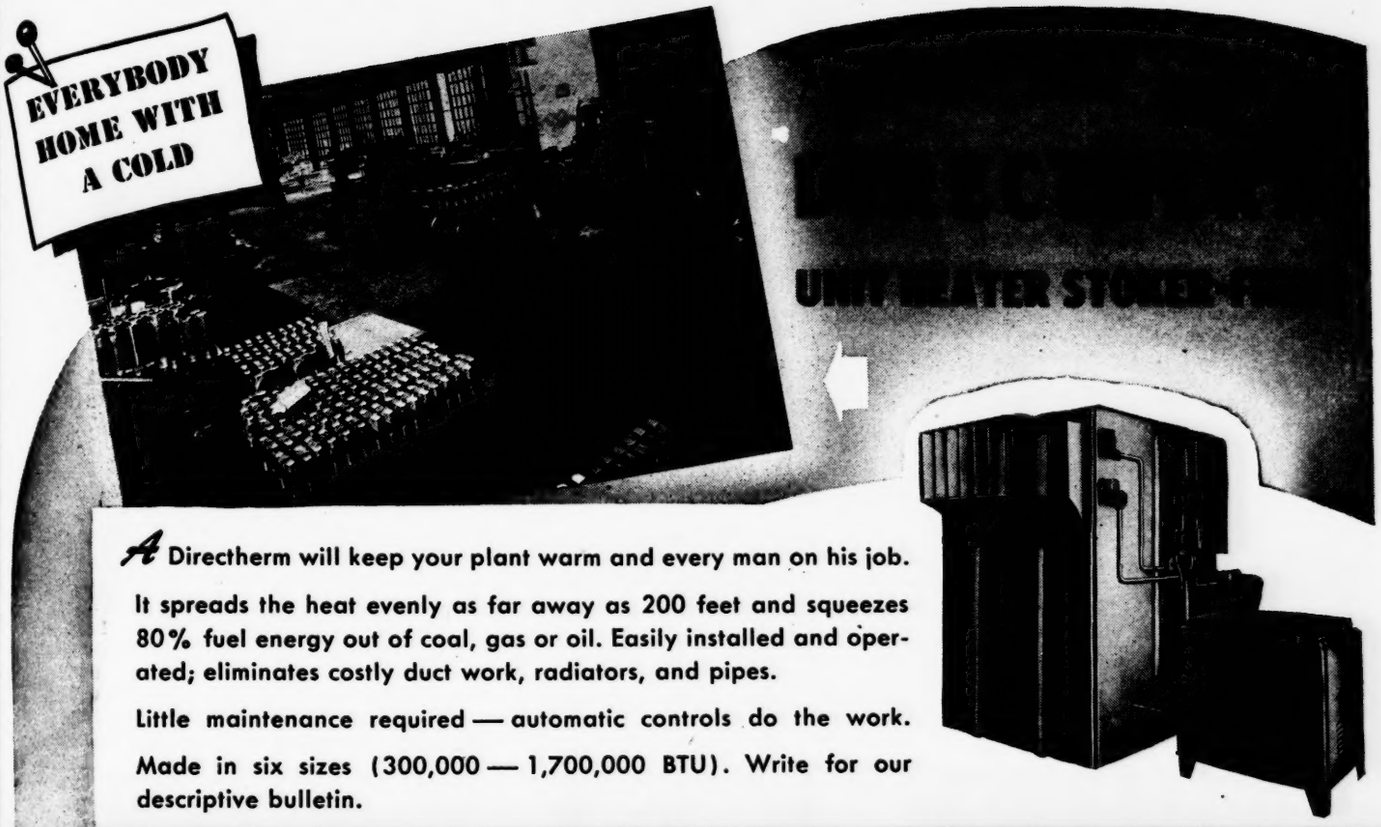
City	Monthly Degree-Days			Cumulative Degree-Days[1]			Year[2]
	Degree-Days, Dec., 1943	Degree-Days, Dec., 1942	Degree-Days, Dec., Normal	Season, 9/1/43-12/31/43	Season, 9/1/42-12/31/42	Season, Normal, 9/1-12/31	Normal, 9/1-6/30
Abilene, Texas	673	535	592	1053	900	925	2061
Albany, New York	1297	1313	1147	2824	2704	2439	6580
Albuquerque, New Mexico	1020	749	924	1908	1497	1827	4298
Alpena, Michigan	1262	1303	1247	3047	2958	2951	8299*
Anaconda, Montana	1249	1129	1238	2959	3064	3171	8357**
Asheville, North Carolina	800	820	837	1839	1656	1689	4232
Atlanta, Georgia	644	670	639	1248	1123	1131	2890
Atlantic City, New Jersey	898	927	893	1798	1617	1735	5176
Augusta, Georgia	527	540	549	967	832	886	2161
Baker, Oregon	1078	1098	1156	2502	2645	2815	7163
Baltimore, Maryland	863	928	874	1726	1611	1664	4533
Billings, Montana	1021	1082	1194	2334	2658	2816	7119
Binghamton, New York	1317	1244	1156	2842	2524	2515	6808
Birmingham, Alabama	679	568	595	1359	942	913	2352
Bismarck, North Dakota	1254	1623	1553	2944	3584	3514	9192
Block Island, Rhode Island	990	1031	884	1975	1979	1736	5788
Boise, Idaho	996	979	1011	2154	2245	2285	5552
Boston, Massachusetts	1097	1132	1026	2220	2169	2130	6045
Bozeman, Montana	1282	1150	1293	2978	3052	3286	8521**
Buffalo, New York	1188	1233	1104	2718	2493	2372	6822
Burlington, Vermont	1424	1424	1287	3083	2998	2773	7514
Butte, Montana	1350	1221	1261	3239	3369	3183	8272
Cairo, Illinois	896	853	822	1707	1481	1499	3909
Canton, New York	1472	1475	1342	3215	3042	2994	8020
Charles City, Iowa	1263	1503	1390	3024	3102	2895	7588
Charleston, South Carolina	467	455	425	805	638	632	1769
Charlotte, North Carolina	655	729	679	1273	1216	1220	3120
Chattanooga, Tennessee	768	722	694	1567	1347	1225	3118
Cheyenne, Wyoming	1092	1031	1144	2717	2776	2889	7466
Chicago, Illinois	1163	1307	1119	2556	2592	1919	5957
Cincinnati, Ohio	1022	1046	927	2078	1984	1796	4684
Cleveland, Ohio	1081	1134	1060	2353	2200	2185	6155
Columbia, Missouri	1103	1043	985	2171	1957	1896	4922
Columbia, South Carolina	572	593	561	1044	910	916	2364
Columbus, Ohio	1066	1109	1017	2200	2095	2020	5398
Concord, New Hampshire	1363	1352	1234	2941	2860	2732	7353
Concordia, Kansas	1087	1130	1050	2215	2189	2019	5315
Dallas, Texas	633	530	567	1034	863	870	2256
Davenport, Iowa	1140	1316	1175	2476	2541	2333	6289
Dayton, Ohio	1125	1105	1014	2387	2099	1947	5264
Denver, Colorado	927	867	1017	2044	2100	2276	5874
Des Moines, Iowa	1165	1314	1215	2579	2563	2370	6384
Detroit, Michigan	1165	1223	1113	2587	2493	2332	6490
Devils Lake, North Dakota	1430	1829	1710	3414	3922	3871	9970
Dodge City, Kansas	1143	1016	1004	2170	1993	1952	5035
Dubuque, Iowa	1175	1379	1243	2694	2758	2534	6790
Duluth, Minnesota	1455	1633	1519	3573	3654	3489	9443
Eastport, Maine	1351	1353	1228	2916	2932	2890	8520**
Elkins, West Virginia	1070	1051	1035	2529	2162	2217	5697
El Paso, Texas	(a)	537	617	(a)	829	1038	2428
Ely, Nevada	1173	1047	—	2773	2848	—	—
Erie, Pennsylvania	1106	1138	1039	2432	2238	2164	6273
Escanaba, Michigan	1299	1412	1344	3170	3199	3138	8771*
Evansville, Indiana	1013	995	862	2049	1876	1545	4244
Fort Smith, Arkansas	779	685	707	1371	1184	1177	3147
Fort Wayne, Indiana	1197	1294	1138	2692	2551	2211	5925
Fort Worth, Texas	636	545	574	1009	857	826	2148
Fresno, California	536	594	577	971	1042	911	2334
Galveston, Texas	334	220	270	518	304	330	1016
Grand Junction, Colorado	967	940	1162	1948	1966	2278	5548
Grand Rapids, Michigan	1155	1195	1147	2606	2492	2397	6535
Green Bay, Wisconsin	1283	1438	1324	3051	3023	2882	7825
Greensboro, North Carolina	799	846	760	1678	1545	1395	3529
Greenville, South Carolina	658	724	738	1315	1223	1334	3380
Harrisburg, Pennsylvania	1011	1067	992	2176	2046	1981	5375
Hartford, Connecticut	1183	1200	1060	2469	2347	2157	6036
Hatteras, North Carolina	597	539	493	1020	767	727	2571
Havre, Montana	1141	1386	1383	2706	3250	3314	8700
Helena, Montana	1293	1155	1245	3031	3027	3058	7898
Houston, Texas	371	270	335	607	416	449	1157
Huron, South Dakota	1256	1506	1431	2909	3245	3085	8004
Indianapolis, Indiana	1066	1140	1017	2229	2121	2002	5298
Kansas City, Missouri	1069	1059	1008	2110	1966	1866	4956
Kewanee, Illinois	1202	1358	1118	2666	2695	2311	6139
Knoxville, Tennessee	782	809	781	1632	1469	1483	3670
La Crosse, Wisconsin	1325	1453	1311	3112	2983	2778	7322
Lander, Wyoming	1384	1162	1419	3038	3083	3075	7947

[1]Cumulative data identical with monthly figures for September only. In subsequent months, cumulative figures will show record for whole heating season to and including month in question.

[2]Figures in this column are normal totals for a complete heating season, September to June, incl.

Figures in this table, with eight exceptions, based on local weather bureau reports. Exceptions are Utica, Lewiston, Kewanee, and Sioux City, figures for which are furnished through the courtesy of Coke Sales Department, Central New York Power Corp., Utica, N. Y.; Norman E. Ross, Bursar, Bates College, Lewiston, Me.; J. M. Hartman, Engineering Department, Kewanee Boiler Corp., Kewanee, Ill., and Alfred R. Wagstaff, Engineer, Sioux City Foundry and Boiler Co., Sioux City, Iowa, respectively; Anaconda, Bozeman, Butte and Livingston, Mont., through the courtesy of the Montana Power Company.

[Table Concluded on page 100



A Directherm will keep your plant warm and every man on his job.

It spreads the heat evenly as far away as 200 feet and squeezes 80% fuel energy out of coal, gas or oil. Easily installed and operated; eliminates costly duct work, radiators, and pipes.

Little maintenance required — automatic controls do the work.

Made in six sizes (300,000 — 1,700,000 BTU). Write for our descriptive bulletin.

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WING REVOLVING UNIT HEATERS

Wing Revolving Unit Heaters insure uniform mixture and circulation of heated air by the use of discharge outlets which revolve slowly and project the streams of heated air to the working level sweeping around in a complete circle. As the moving air circulates along the working level, it mixes uniformly with the cold air, circulates around heavy machinery and obstructions and penetrates into cold, damp corners—insuring a comfortable uniform temperature throughout the entire plant. These Wing Revolving Heaters are suitable for all types and sizes of buildings because the deflectors on the discharge outlet can be adjusted to propel the air in two different concentric circles which permits coverage of a great area.

L. J. Wing Mfg. Co.

158 West 14th St., New York, N. Y. — Factories: Newark, N. J.

Degree-Days for December, 1943 (Concluded)

HEATING AND VENTILATING'S 16th year of publication of monthly degree-day data

City	Monthly Degree-Days			Cumulative Degree-Days[1]			Year[2]
	Degree-Days, Dec., 1943	Degree-Days, Dec., 1942	Degree-Days, Dec., Normal	Season, 9/1/43-12/31/43	Season, 9/1/42-12/31/42	Season, Normal, 9/1-12/31	Normal, 9/1-6/30
Lansing, Michigan	1231	1280	1209	2893	2746	2665	7048
Lewiston, Maine	1494	1346	1295	3154	2946	2837	7707
Lincoln, Nebraska	1093	1225	1132	2360	2391	2201	5999
Little Rock, Arkansas	759	679	651	1359	1184	1079	2811
Livingston, Montana	1080	1015	1117	2479	2785	2731	7205
Los Angeles, California	255	209	301	356	354	424	1504
Louisville, Kentucky	928	960	849	1841	1758	1587	4180
Lynchburg, Virginia	778	865	800	1593	1539	1557	3980
Macon, Georgia	565	518	567	1084	847	897	2201
Madison, Wisconsin	1253	1454	1321	2910	2944	2758	7429
Marquette, Michigan	1287	1379	1302	3155	3172	3054	8693*
Memphis, Tennessee	755	704	663	1443	1175	1127	2950
Meridian, Mississippi	574	491	549	1114	829	883	2160
Milwaukee, Wisconsin	1238	1391	1221	2859	2865	2601	7245
Minneapolis, Minnesota	1287	1535	1404	3069	3218	2941	7850
Modena, Utah	1083	964	1168	2394	2382	2672	6562
Montgomery, Alabama	511	446	496	958	713	769	1884
Nantucket, Massachusetts	987	1016	905	1970	1975	1912	5957
Nashville, Tennessee	816	791	744	1613	1391	1363	3507
New Haven, Connecticut	1111	1130	1017	2320	2131	2109	5895
New Orleans, Louisiana	349	255	301	608	378	403	1024
New York, New York	1008	1034	955	1997	1844	1849	5347
*Nome, Alaska	1269	1570	1509	3073	3276	3379	14580**
Norfolk, Virginia	691	701	685	1290	1097	1195	3350
North Head, Washington	594	633	654	1538	1711	1785	5452**
North Platte, Nebraska	1051	1049	1173	2354	2424	2486	6366
Oakland, California	455	481	496	989	1089	1099	3143**
Oklahoma City, Oklahoma	873	753	815	1460	1358	1379	3613
Omaha, Nebraska	1150	1264	1175	2516	2485	2284	6131
Oswego, New York	1232	1266	1138	2723	2540	2494	7088
Parkersburg, West Virginia	1014	1016	921	2130	1903	1830	4775
Peoria, Illinois	1204	1255	1150	2569	2469	2323	6109
Philadelphia, Pennsylvania	925	1000	905	1827	1760	1735	4855
Phoenix, Arizona	351	281	425	485	376	581	1405
Pittsburgh, Pennsylvania	1021	1054	967	2524	2003	1949	5235
Pocatello, Idaho	1125	1078	1181	2488	2546	2649	6655
Portland, Maine	1366	1297	1159	2926	2856	2605	7218
Portland, Oregon	712	645	729	1514	1488	1724	4469
Providence, Rhode Island	1073	1126	1026	2173	2142	2130	6015
Pueblo, Colorado	1087	909	1029	2370	2084	2164	5514
Raleigh, North Carolina	723	744	694	1405	1252	1253	3234
Rapid City, South Dakota	1096	1149	1172	2555	2781	2701	7118
Reading, Pennsylvania	992	1070	1001	2080	1995	2014	5389
Redding, California	467	615	570	867	1054	919	2451
Reno, Nevada	1087	878	973	2341	2129	2284	5892
Richmond, Virginia	797	837	763	1565	1449	1380	3695
Rochester, New York	1232	1260	1125	2767	2575	2414	6732
Roseburg, Oregon	769	611	719	1575	1424	1741	4428
Roswell, New Mexico	859	612	808	1576	1193	1510	3484
Sacramento, California	524	588	574	933	1010	996	2653
St. Joseph, Missouri	1114	1138	1079	2247	2142	1960	5161
St. Louis, Missouri	1006	982	936	1918	1787	1738	4585
Salt Lake City, Utah	1045	984	1020	2154	2209	2149	5555
San Antonio, Texas	453	314	347	760	506	473	1202
San Diego, California	217	236	288	360	389	487	1645
Sandusky, Ohio	1123	1187	1063	2373	2286	2142	6208
San Francisco, California	356	399	428	760	959	949	3264**
Sault Ste. Marie, Michigan	1435	1515	1370	3443	3445	3247	9285**
Savannah, Georgia	403	388	397	686	545	598	1490
Scranton, Pennsylvania	1167	1190	1085	2555	2361	2274	6129
Seattle, Washington	681	638	719	1589	1630	1869	4934**
Sheridan, Wyoming	1073	1104	1320	2633	2818	3198	8008
Shreveport, Louisiana	561	524	499	1019	809	769	1938
Sioux City, Iowa	1210	1397	1265	2761	2882	2583	6898
Spokane, Washington	1107	1011	1057	2493	2483	2583	6355
Springfield, Illinois	1129	1150	1039	2224	2157	2002	5373
Springfield, Missouri	1010	904	893	2032	1779	1674	4428
Syracuse, New York	1248	1282	1172	2794	2600	2491	6893
Tacoma, Washington	758	664	756	1801	1754	1951	5181**
Terre Haute, Indiana	1066	1105	992	2170	2055	1840	4872
Toledo, Ohio	1206	1224	1079	2655	2419	2189	6077
Topeka, Kansas	1066	1071	1008	2085	1982	1883	4969
Trenton, New Jersey	1002	1066	930	2073	1947	1760	4933
Utica, New York	1354	1310	1144	2986	2728	2537	6796
Valentine, Nebraska	1142	1172	1245	2641	2778	2725	7039
Walla Walla, Washington	989	810	896	2077	1801	1909	4808
Washington, Dist. Columbia	885	937	896	1778	1653	1741	4626
Wichita, Kansas	1029	1006	955	1911	1877	1783	4673
Williston, North Dakota	1297	1564	1591	3040	3578	3658	9323
Winnemucca, Nevada	1042	973	1080	2262	2297	2589	6427**
Yakima, Washington	1004	1082	1091	2194	2297	2428	5599

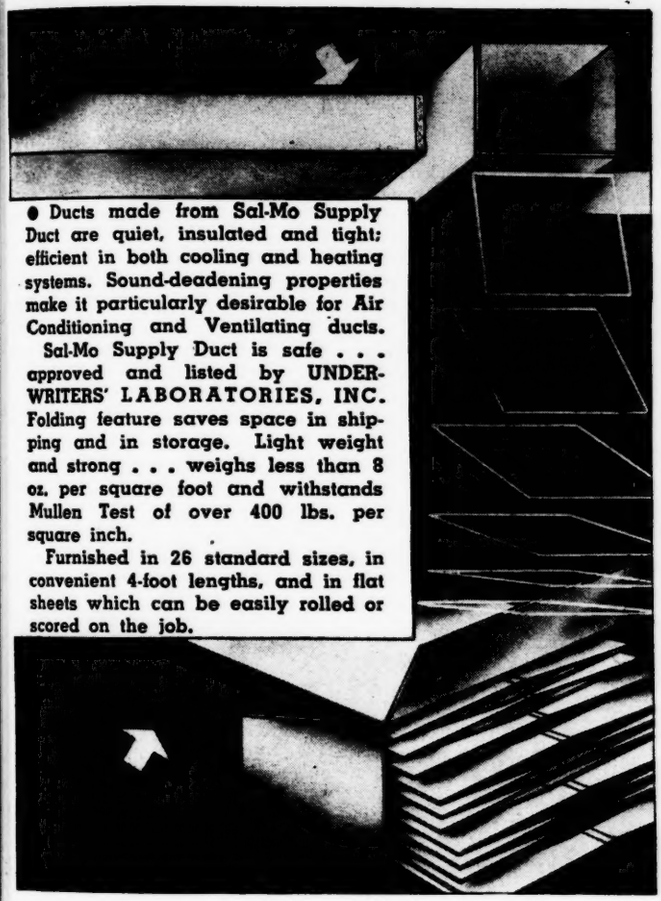
*Includes August. **Includes July and August. (a) Data not yet available.

[1]Cumulative data identical with monthly figures for September only. In subsequent months, cumulative figures will show record for whole heating season to and including month in question.

[2]Figures in this column are normal totals for a complete heating season, September to June, incl.

For Air Conditioning - Ventilating
Warm Air Heating . . .

SALMO Supply Duct



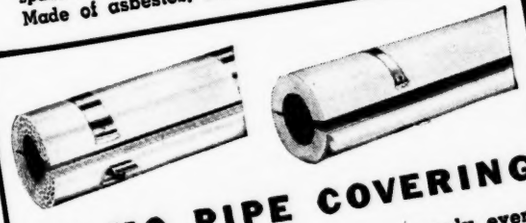
● Ducts made from Sal-Mo Supply Duct are quiet, insulated and tight; efficient in both cooling and heating systems. Sound-deadening properties make it particularly desirable for Air Conditioning and Ventilating ducts.

Sal-Mo Supply Duct is safe . . . approved and listed by UNDERWRITERS' LABORATORIES, INC. Folding feature saves space in shipping and in storage. Light weight and strong . . . weighs less than 8 oz. per square foot and withstands Mullen Test of over 400 lbs. per square inch.

Furnished in 26 standard sizes, in convenient 4-foot lengths, and in flat sheets which can be easily rolled or scored on the job.

SAL-MO NO. 77 ASBESTOS DUCT BOARD

An asbestos product developed for constructing ducts in air conditioning, ventilating and warm air heating systems. Sal-Mo Ductboard is light in weight, easily and quickly applied, moisture-resistant. Available in sheets 33" x 48", just the right size to cover two joist openings spaced at 16" centers, making two cold air return ducts. Made of asbestos, No. 77 Ductboard is fireproof.



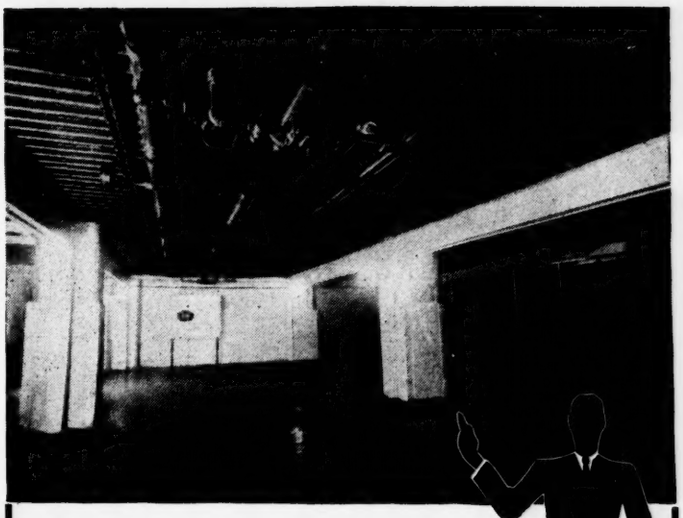
SAL-MO PIPE COVERING

There is a Sal-Mo Pipe Covering for nearly every known purpose. Aircell, Multicell and Sixcell for steam and hot water lines; Wool Felt Covering asbestos paper lined for hot water or asphalt paper lined for cold water pipes. Sal-Mo Alaska Frost-Proof for protecting water and gas pipes exposed to low temperatures and Sal-Mo Anti-Sweat for preventing damage from sweating, dripping pipes.

Asbestos

SALL MOUNTAIN COMPANY

176 WEST ADAMS STREET • CHICAGO 3 • ILLINOIS



WHITLOCK

REFRIGERATING EQUIPMENT

WHITLOCK ENGINEERS KNOW HOW

The experience of Whitlock engineers, covering a wide range of applications over a period of years, provides the properly designed refrigerant equipment required for your particular needs.

WHITLOCK HAS THE FACILITIES

Whitlock's staff, expert in the design of condensers, receivers, welded headers, prefabricated piping, coils and allied products, combined with Whitlock's extensive fabricating equipment—heat treating furnace for the relief of welding stresses, x-ray machine for examining material soundness—provide the high quality product you demand.

WHITLOCK HAS THE MATERIAL

An adequate supply of medium and heavy steel plate, forged steel pipe flanges, and steel pipe insures that, with a suitable priority rating, Whitlock can make prompt delivery of prefabricated units.

The **WHITLOCK MANUFACTURING CO.**

40 South St., Elmwood, Hartford 1, Conn.

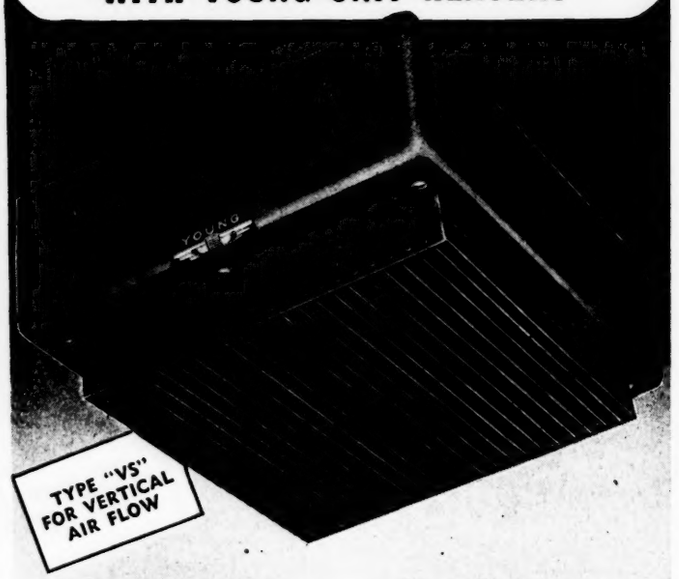
BRANCH OFFICES: New York, Boston, Chicago, Philadelphia, Detroit, Richmond. Authorized representatives in other principal cities —consult local telephone directory under our name.

Whitlock Equipment (Whitlock-Darling) is manufactured in Canada by Darling Brothers, Ltd., Montreal.

DISCOMFORT DEFEATS PEAK PRODUCTION

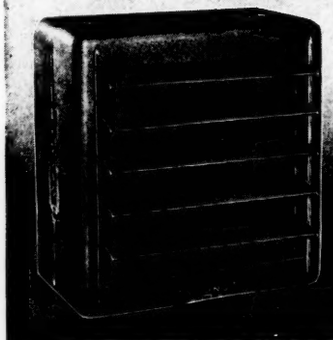
WARM UP *the* COLD SPOTS

WITH YOUNG UNIT HEATERS



More consideration than ever will be given to the comfort and health of war workers this winter . . . not only to prevent illness that results in absenteeism, but to invite employment, build loyalty and morale, and increase production efficiency. If heating facilities were inadequate last year, if cold spots were detected or if departments have been rearranged, Young Streamaire Unit Heaters can be used to advantage. Their saving of floor space, ease of installation, low first cost, efficient operation and low maintenance cost make Young Unit Heaters desirable for existing or new buildings and additions. Write us direct or consult with the Young representative in your locality, for the solution to heating and air conditioning problems.

YOUNG RADIATOR COMPANY
Dept. 144-B Racine, Wis., U.S.A.



TYPE "SHS" FOR HORIZONTAL AIR DELIVERY

BUY BONDS
PRODUCE MORE
SALVAGE SCRAP
WIN THE WAR

YOUNG

HEAT TRANSFER PRODUCTS

Oil Coolers - Gas, Gasoline, Diesel Engine Cooling Radiators - Intercoolers - Heat Exchangers - Engine Jacket Water Coolers - Unit Heaters - Convectors - Condensers - Evaporators - Air Conditioning Units - Heating Coils - Cooling Coils - and a complete line of Heat Transfer Equipment for aircraft.

Getting Personal

Irwin R. Hoffman

It is rare to find an engineer with the coal-burning experience possessed by **Irwin R. Hoffman** (How to Burn Coal for Large Buildings, page 51).

He received his mechanical engineering degree from Drexel Institue of Technology in 1922 and for the next two years was cadet engineer with the United Gas Improvement Co., Philadelphia. From 1924 to 1931 he was erection engineer for McClave-Brooks Co., serving part of the time as Chief Engineer at the New York office.

Since 1931 he has been Chief Engineer, Central Coal Co., New York City. He introduced the use of blended fuels in large buildings, and has promoted a greater use of bituminous coal in this anthracite area.

Mr. Hoffman cooperated with the New York Ordnance Department in the study of oil and pulverized coal mixtures, and he has worked with the Petroleum Administration for War in making large conversions from plants burning oil to coal. He has both lectured and written extensively on the proper burning of coal.

Mr. Hoffman is serving as consulting fuel engineer for a number of coal companies in New York City.

Confidentially, outside of office hours, he is an amateur landscape painter and has had a number of his oil paintings exhibited in galleries.

George Elmer May

George Elmer May, (Foreword, Gas for Summer Air Conditioning, page 55) Utilization Engineer, New Orleans Public Service, is a true native of the colorful city of New Orleans, La. Like his parents, he was born in that much publicized city.

He graduated in both Mechanical and Electrical Engineering from Tulane University. Incidentally, he recently equipped the handsome new Tulane library with gas air conditioning, designing the system both for the readers' comfort and for the proper preservation of the books and manuscripts.

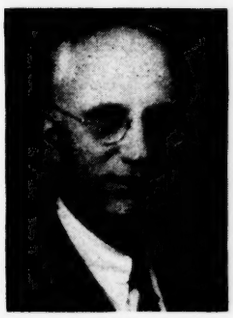
His entire business career has been with the New Orleans Public Service.

Mr. May is well known in the heating and ventilating field. He organized and was first president of the Delta (Louisiana) Chapter of the American Society of Heating and Ventilating Engineers. He is chairman of the American Gas Association Advisory Subcommittee on Gas Summer Air Conditioning Research, and he is ex-officio member of the Joint Committee on Gas Summer Air Conditioning of the Industrial and Commercial Gas Section and the Residential Gas Section of the AGA. He has written many papers on air conditioning.

Both Purdue and the University of Tennessee must share the honor for the engineering education provided **Frank Earl Markel** (Radiant Heat Installed in Chapels at Naval Air Station, page 82). He is a native of Indiana.

He started in construction work in 1904 and has been closely associated with the field ever since. In 1918

he became affiliated with war work and joined the office of H. Koppers Co., Pittsburgh, as designer. He remained there until World War I was over and returned to his home in Atlanta, Georgia. Mr. Markel opened an engineering office there for the handling of mechanical design and continued in that work until July, 1940, when he joined the office of Robert & Company. He moved to Jacksonville to help in the Navy's building program.



Frank Earl Markel

During the past three and a half years he has handled the design work on 18 Navy bases and three Army bases. Mr. Markel said "In peacetime this would keep an office busy for a generation."

AGITAIR PRODUCTS

Air Devices, Inc., 17 E. 42nd St., New York 17, N. Y.

A 12-page standard size bulletin entitled "Agitair Products," illustrating and describing the products made by Air Devices and including air diffusers, heavy-duty ranges, bake ovens and stoves, air exhausters, air filters, hot water generators, heat baffles for oil burners, hot gas generating furnaces.

FACTS REGARDING COAL

Bituminous Coal Institute, 60 E. 42nd St., New York 17, N. Y.

This new publication, "Coal Facts" shows why coal is imperative towards winning the war, and necessary for industry and home use. A large amount of data is presented including coal consumption, mine safety and wage records.

THERMOMETERS

Johnson Service Company, Milwaukee 2, Wisc.

A standard size 16-page catalog bulletin No. 20, printed in color and including information on, illustrations of, and data applying to the company's line of thermometers.

MOTORS

Century Electric Company, St. Louis 3, Mo.

Bulletin 6-1, devoted to the company's Form J splash proof motors in sizes 1½ to 15 hp, which are now available with two-way ventilation. Cooling air is drawn in through baffled openings in the bottom of the end brackets by a fan on each end of the rotor, blown through and around into all parts of the motor; and out through louvered openings in the side of the frame below the center line and at the bottom. This construction protects the vital parts of the motor during plant wash down or for outdoor operation.

PIPE TOOLS

American Pipe Tool Co., 4856 W. Kinzie St., Chicago.

A 3⅝ x 6¼ in., 40-page catalog of the company's line of pipe wrenches, vises, reamers, stocks, dies, and cutters.

(Concluded on page 104)

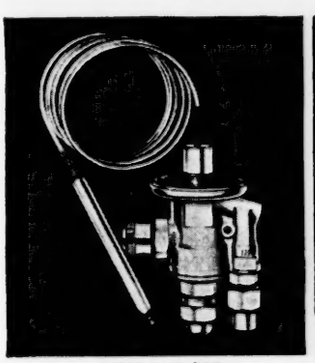
▸Ask your service man to place an A-P "TRAP-DRI" System Protector ahead of each expansion valve, regardless of make. This stops all dirt, acids, solder particles and moisture that may otherwise cause valve trouble and service expense.

▸Then be sure he checks all expansion valves, replacing sluggish and inaccurate ones with A-P DEPENDABLE Thermostatic Expansion Valves. It's one way of preventing costly service shutdowns later, and assuring a smooth-operating system.

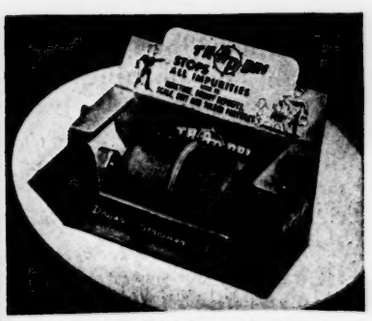
▸This is *preventative service*, costing little now, that may save you money later, and provide greater efficiency from your display cases, storage units, and other refrigeration and air conditioning systems.

DEPENDABLE REFRIGERANT VALVES

Thermostatic Expansion — Solenoid — Constant Suction Pressure — Water Regulating Valves — "Trap-Dri" System Protectors — Water Solenoids — Cooling Thermostats . . stocked and sold at refrigeration jobbers everywhere—recommended and used by leading refrigeration service engineers.

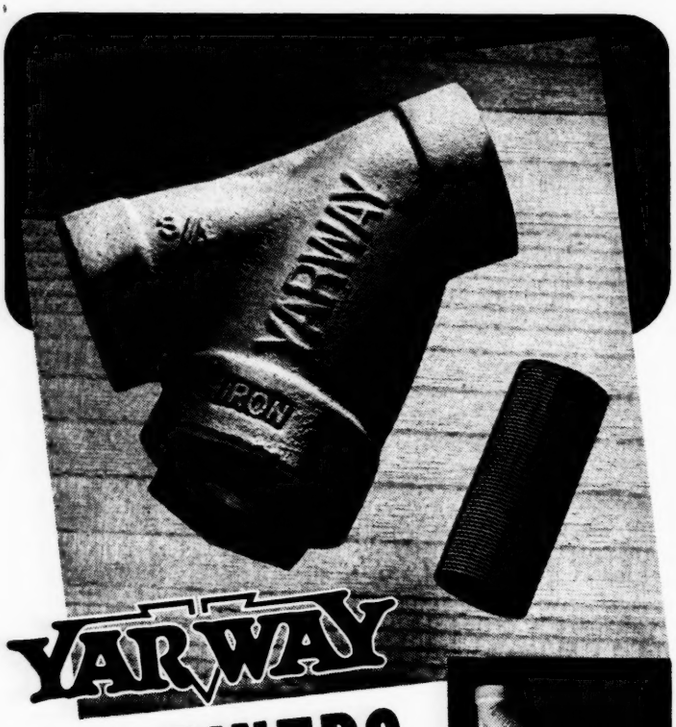


A-P Model 205
Thermostatic Expansion Valve



A-P "Trap-Dri" System Protector
Stops all impurities and moisture

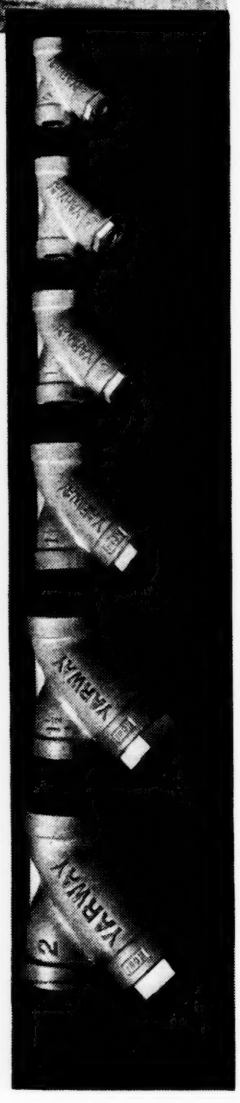
AUTOMATIC PRODUCTS CO.
2462 North Thirty-Second Street • Milwaukee 10, Wisconsin
Export Department: 15 East 40th Street • New York 16, N. Y.



YARWAY

STRAINERS

"Police the Pipelines"



YARWAY STRAINERS are *selling by the thousands* because they are better engineered for the service.

"*The Screen is the Thing*"—a high-grade, woven Monel wire screen that stops the dirt, lets fluids flow freely.

Then, too, purchasers like the *body finish—Cadmium plating inside and out* for protection against corrosion.

And last but not least, it is "*Easy to Clean*" having a steel blow-off bushing, precision machined with straight thread. Screen and bushing come out together—go back together, automatically aligning.

Six sizes, ½" to 2" for pressures up to 600 lb serve practically all strainer needs.

Sold by over 100 Mill Supply Houses. See your Supply House or write for Bulletin S-200.

YARNALL-WARING COMPANY
104 Mermaid Ave. PHILADELPHIA 18, PA.

YARWAY STRAINERS

(Concluded from page 103)

UNIVERSAL JOINT

Condenser Service & Engineering Co., 95 River Street, Hoboken, N. J.

Bulletins CS-10 describes precision cut, manganese-bronze, bevel-gear universal joint with angular range of 0° to 135° on vertical center line or 0° through 360° on horizontal center line. For remote control of valves, pumps, engines, antennae, ventilators and other equipment in inaccessible locations, or for centralized control of widely separated units. Bulletin also gives data on brackets and shaft hangers.

HELMETS FOR SOLDIERS

McCord Radiator & Mfg. Co., Detroit.

A booklet of 24 pages with many illustrations presents "The Story of the New American Helmet." It shows the wide use of this helmet by the American soldier in field service and explains the part that the company has had in the development of the helmet.

CONTROLS

Worner Electronic Devices, 848 N. Noble Street, Chicago 22, Ill.

Bulletin 9143, covering operation of photo-electric devices for automatically controlling equipment for production, safety, lighting, combustion, sabotage, burglary, etc. Written in non-technical language.

SOLENOIDS

L. M. Persons Corp., 6301 Manchester Ave., St. Louis 10, Mo.

Two single page loose-leaf-punched pages for the company's catalog. One sheet is devoted to types 1000-1E and 1000-2E aircraft solenoid valves. The other gives specifications of the Persons solenoid series 750, 750A, 751, 751A, and 765-1E. Specifications, illustrations, dimensional data, and performance curves are included.

WITH THE MANUFACTURERS

THE MARLEY CO., INC.

Kansas City, Kan., has established a new divisional office at 2006 Esperson Bldg., Houston, Tex., to provide immediate and direct sales and service coverage of the Texas Gulf Coast, Northwestern Louisiana and Southwestern Arkansas. The new office adds its increased facilities to those in Dallas, Corpus Christi, San Antonio and El Paso where the Company's representation remains unchanged. Ray T. Jenkins, formerly in charge of the Tulsa divisional office, heads the Houston division. Mr. Jenkins joined the Marley organization in 1933 and represented the Company in Dallas for three years before assuming direction of the Tulsa territory in 1937.

YORK-SHIPLEY, INC.

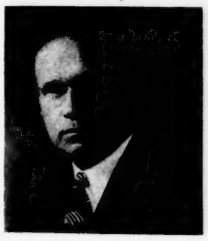
York, Pa., effective December 31, 1943, took over all of the assets and liabilities of the York Oil Burner Company, Inc. York-Shipley was the owner of all of the stock of York Oil Burner Company, Inc., and the change is one in corporate name only.

THE VIBRATION ENGINEERING CO.

521 Fifth Ave., New York, has taken over the business formerly conducted by B. T. Butterworth, Jr. Mr. Butterworth heads the company as president, Harry Hayden is secretary and treasurer. The company will expand its activities in the industrial field.

THE WEATHERHEAD COMPANY

Cleveland, Ohio, has created a position of executive vice-president, to be filled by H. I. Lewis, Hartford, Conn., formerly of the American Hardware Company. Mr. Lewis, a mechanical engineer, was also formerly with the Seybold Division of Harris-Seybold-Potter, and with American Type Founders. Henry F. Bailey, Cleveland banker, will take over the position of vice-president in charge of finance. Mr. Baily has been with the National City Bank of Cleveland for seven years, was formerly with the Federal Reserve Bank and prior to that with Longwood Greene Engineers, Inc. He is a registered professional engineer.



H. I. Lewis

SARCO COMPANY, INC.

announces the formation of an associate company to be known as Sarcotherm Controls, Inc., to market the company's weather controls for hot water and radiant heating as well as other heating system controls. Headquarters of the new company will be in the Merchandise Mart, 222 North Bank Drive, Chicago 54, Ill. E. M. Mittendorff, of Sarco Company's Chicago office, has been elected vice-president of the new company and will act as its general manager.

MINNEAPOLIS-HONEYWELL REGULATOR CO.

has announced the purchase of a 10-story building from the Ford Motor Company in order to increase facilities of the Minneapolis-Honeywell organization by 50%. While intended as a post-war development, the building will be used to increase the plant production and storage facilities which are now distributed among seven leased warehouses.

H. A. THRUSH & CO.

Peru, Ind., has appointed Nelson L. Little, as representative in Richmond, Va., to cover the Virginia - North Carolina territory. Mr. Little was formerly in the engineering department at Peru as well as field representative in the New England territory.

YOUNG RADIATOR CO.

Racine, Wis., has appointed W. P. Nevins Company as sales and engineering representatives in the Chicago territory, including Lake and Cook counties in Illinois and Lake and Porter counties in Indiana. Engineering associates of Mr. Nevins, who will work with him in the territory, are C. L. Carlson and F. L. Winkler.

TITEFLEX, INC.

is the new name of the Titeflex Metal Hose Co., Newark, N. J. The change of name has been made with the realization that Titeflex Metal Hose Co. is no longer descriptive of all the products manufactured by the company. After the war this will be even more true, as Titeflex, Inc. anticipates the manufacture and sale of a broader line of products to a wide range of industries. However, as a pioneer manufacturer of flexible metal hose, it is expected that metal hose will always be an important part of the company's production.

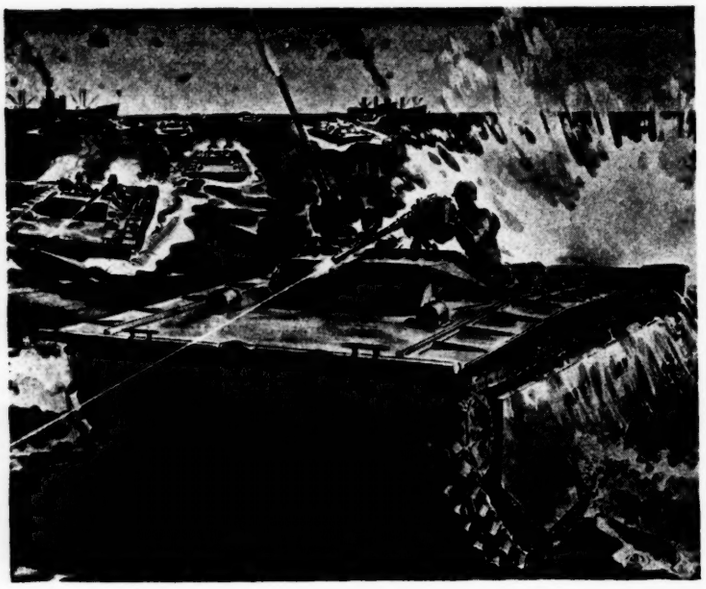
ALLEN-BRADLEY CO.

Milwaukee, Wis., has appointed Rietze & Co., 1017 E. Broadway, Louisville 4, Ky., as sales representatives for the Allen-Bradley line of electric controlling apparatus. Mr. Rietze will handle the southern Indiana and western Kentucky territories.

Attractive in appearance . . . Efficient in performance . . . Rigid one-piece construction . . . Easily and quickly installed . . . Large range of types and sizes to meet every Heating, Ventilating and Air Conditioning problem.

TUTTLE & BAILEY
INCORPORATED
NEW BRITAIN, CONN.

Send for the newest Aerofuse Outlet Engineering Guide and Catalog.



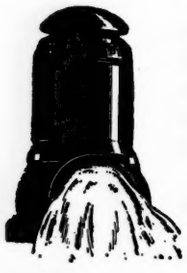
The vicious land and sea fighter

—the amphibious

WATER BUFFALO

Designed by the Engineers of
PEERLESS PUMPS

America's newest fighting fortress has gone to War with a vengeance. It's the Water Buffalo. It swims through sea and surf, it crawls over beachheads, up over rocky terrain, trundles through mud and swamps—it goes everywhere, loaded with devastation. Peerless engineers designed this deadly attacker. Their "know how" of engineering hydraulics —their everyday knowledge of water problems and their detailed knowledge of pumps, were largely responsible for Peerless engineers becoming the world's authority on amphibian tractor designs.



Needless to say, Peerless engineering forces have been greatly strengthened and have knowledge of real value to the pump industry.

First on the Waterfront—
at War and at Home

PEERLESS PUMP DIVISION
Food Machinery Corporation

★ WATER OR OIL LUBRICATION

Peerless
TURBINE PUMPS

PERFORMANCE WITH ECONOMY

Factories: 301 W. Ave. 26, Los Angeles 31, Calif.
Canton 6, Ohio; San Jose 5 and Fresno 16, Calif.

PERSONALS AND PERSONNEL

GENE WEDEREIT

has been appointed advertising manager, Tube Turns and the Girdler Corporation, Louisville, Ky. He was formerly vice-president, and director of creative service with the E. E. Schmidt Co., Milwaukee. He came to Louisville as a result of his work in the designing and production of Tube Turns' 240 page catalog. Walter Girdler, Jr., formerly advertising manager, is now director of personnel.

Gene Wedereit

JAMES A. DWYER

has been appointed general manager of sales and branches of Crane Co., with headquarters in Chicago, effective January 1. Mr. Dwyer joined the Crane organization in 1917 at the Philadelphia branch. He has been successively estimator, salesman, chief clerk, branch sales manager, assistant branch manager, branch manager, and district manager. About a year ago he was made manager of branch houses, which responsibilities he will continue to assume in addition to his new duties as general manager of sales.

WILLIAM R. NORTHLICH

previously with the Washington, D. C., office of Owens-Corning Fiberglas Corp., has been transferred to the company's Toledo, Ohio, office. He affiliated with the company following his graduation. After his graduation from the University of Illinois in 1927, he went with the U. S. Gypsum Company.

Before joining Fiberglas, he was with the Buchen Company, a Chicago advertising agency. He is a native of Kansas City.

William R. Northlich

How to Burn Coal

(Concluded from page 51)

fire where it will burn with very little smoke. Fresh coal is placed on the dead plate and the process is repeated.

Where steady steaming is required, the spreading method is more efficient. The fire is built up to a bed 8 to 12 in. thick. About half the fire is covered at a time. After this charge is ignited, coal is spread on the other half to insure burning gases. It is important to fire alternately in this manner to minimize smoke. Caking is controlled either by raking the crust or slicing the fire. Raking the surface is best done with a three-pronged fork, while the slice bar is used by running it along the top of the grates and easily raising the coke, just enough to crack and loosen the fire.

This operation is performed on one portion of the fire at a time before the balance of the fire is covered with coal. Smoke will only result from the side that is broken and comparatively little from the fresh coal. At the same time, the loosened side of the fuel bed will accelerate steaming.

Bituminous coal can also increase the capacity of a boiler as it has a higher heating value than anthracite. In terms of horsepower, one sq ft of grate surface can easily produce about 4 hp with soft coal and 3 hp with anthracite. In terms of radiation load, one sq ft of grate can comfortably carry about 560 sq ft of radiation with bituminous and 420 sq ft with hard coal. These figures are nominal and can be increased by intelligent firing, and by the utilization of stronger drafts and special equipment. It is well to check boiler grate area with radiation load when difficulty with heating is encountered.

Soft coal is used much more economically with mechanical stoking. With stoking the coal is fed in small quantities at a rate which is coordinated with the steam demand. Gas is distilled off slowly and must pass through the hot fuel bed, resulting in smokeless operation.

Plunger, or ram-feed, stokers can use a wide variety of coals. This also holds where screw-feed stokers are amply sized, except for coal which must be small enough to pass through restricted feeding apparatus. If necessary, any size coal may be used, but large lumps must be broken, usually by striking with the flat of a shovel.

All progressive coal companies maintain engineers to assist consumers in the proper methods of conserving coal. These men are trained in the proper use of many types of coal. A good way to think about the subject is in the words of one of these men: "To burn less coal, burn all of it."

IRRADIATION AS A HEALTH FACTOR

(Concluded from page 50)

Summarizing, the heating of the human body by means of radiant energy has certain physiological effects which are not found with convective heating. Present data suggest that the effects of radiation from sources of moderate temperature are beneficial to health, but the evidence is insufficient to warrant any general conclusion.

Further, only those parts of the body in view of the source are receivers of radiant energy and, in consequence, radiant heating may lead to considerable variations in temperature over different parts of the body surface. Possibly the adverse effects on health of the chilling factor may exceed the benefits derived from the irradiation. No investigations of these effects have yet been noted and until such information is available the utmost caution should be used in prophesying the net effects of radiant energy on health.

FACTS ON TEMPERATURES

In all operations where temperatures require frequent observation, Motoco Indicating Thermometers are dependable aids in conserving power; in preventing waste of time and materials; in safeguarding valuable equipment and insuring its best performance.

Motoco Thermometers are the development of many years experience in manufacturing dial-type instruments for measuring and recording. Their distinctive design eliminates screws, levers, gears, hairsprings and other delicate parts subject to wear and adjustment. Due to simple, strong construction Motoco Indicating Thermometers give long and valuable service.

They are available with rigid stem or flexible capillary tubing for distant reading. Information on models, sizes and prices on request.

Illustrated below are some of the many types of Motoco precision instruments. Left: Aviation Engine Gauge Unit. Center: Recording Thermometer. Right: Pressure Gauge used in oxy-acetylene welding.

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